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ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. XVIII No. 2 FEBRUARY, 1957

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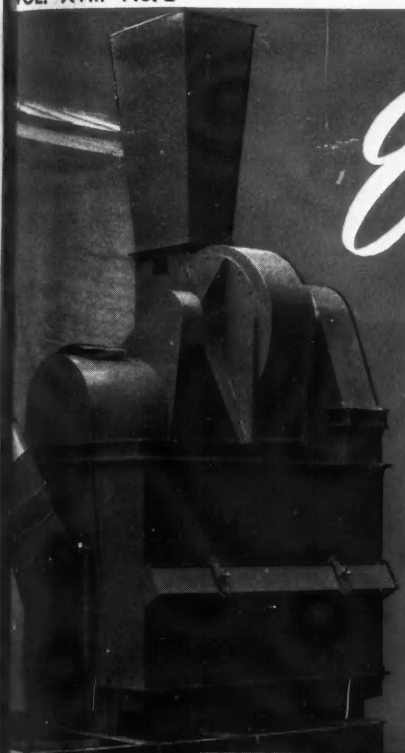

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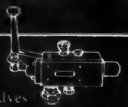

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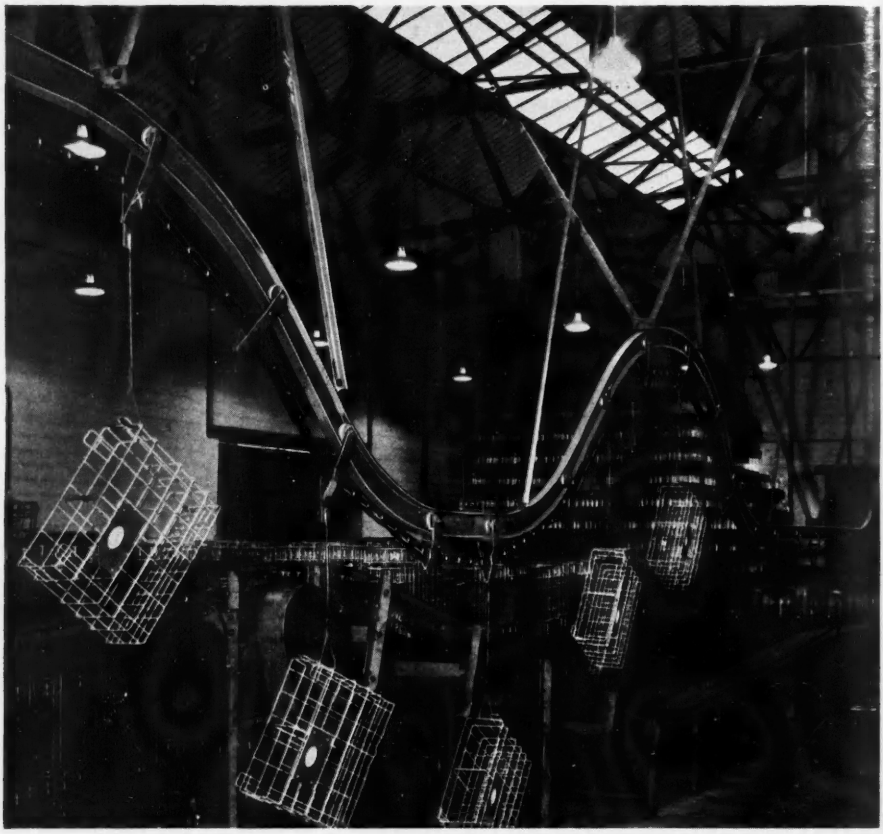

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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

• *RADICALLY NEW CATHODE-RAY TUBE*

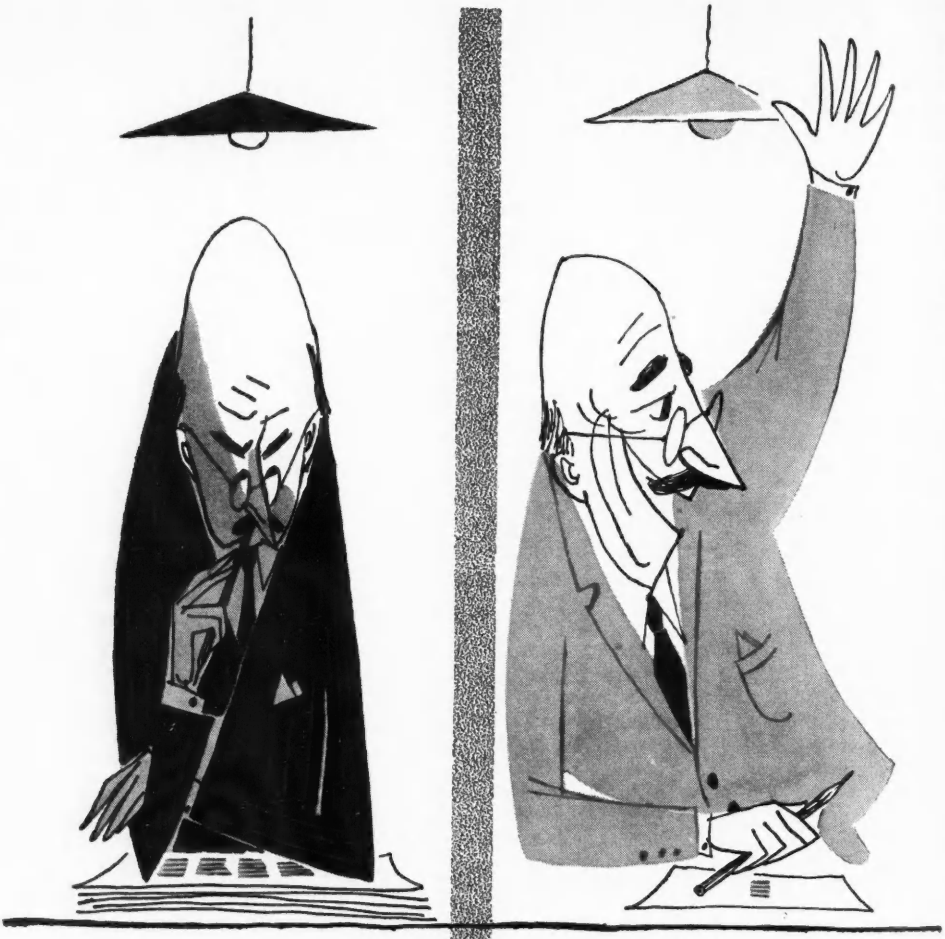
A radically new type of cathode-ray tube for radar, television, and other electronic display applications has been developed, and is claimed to represent a significant advance in the simplification, reliability, and versatility of many types of electronic equipment. The new tube combines most of the essential functions of a microwave receiving set in a single tube envelope, eliminating many of the valves and components required by conventional receivers. In a radar receiver utilizing the new tube, microwave signals pass directly from the antenna into the tube, where, in a single envelope, the signals are amplified, detected, and displayed on the fluorescent screen of the tube. Compared with a conventional radar receiver, this means that the local oscillator, mixer, intermediate frequency amplifier, detector, video amplifier, and their associated circuitry are eliminated. An important feature is the wide selection of channels possible with this tube, which operates over a microwave frequency band of 2000 to 4000 megacycles. The present model of tube is just under 2 ft in length, with a screen diameter of 5 in., but there is no practical limitation to the size of tube face or type of screen, both of which would be determined by the required application.

• *INFRARED SPECTROSCOPY OF LARGE PARTICLES*

The infrared spectra of relatively large particles can now be observed by means of a recently developed technique involving the suspension of particles in a medium having approximately the same index of refraction as the particles themselves. From the resulting spectrum, changes in molecular structure can be determined. Infrared absorption spectroscopy is being extensively utilized for the analysis of large molecules such as those of rubber, plastics, and petroleum constituents. Infrared analysis techniques are also of value in correlating the molecular structure of these materials with their properties. Solid-phase spectroscopy has required the particles analysed to be smaller than the wavelength of incident radiant energy, in order to minimize scatter. Consequently, plastic substances are usually studied as thin films, cast from solution or pressed with application of heat. However, solvents often remain in the plastics and interfere with their spectra, while the application of heat is also unsuitable in some cases, as it can cause thermal degradation. Direct attempts to reduce the plastic particles to the size required by grinding can also result in polymer breakdown. In the new method, the desired result, i.e., the elimination of scatter, is achieved by matching the refractive indices of the particle and of the suspension medium. A typical plastic studied by this method was polyvinyl chloride. As all liquids of the proper refractive index as suspension media for this material exhibit infrared absorption bands, and a complete spectrum can be obtained only by using several suspension media, a special technique, termed the alkali-pellet technique, was subsequently evolved, using pellets composed of a mixture of potassium bromide and polyvinyl chloride. Finely ground and dried potassium chloride was mixed with 0·028 gm of polyvinyl chloride in a dry box and loaded into a die, air being evacuated from the die until a hard vacuum was obtained, after which the assembly was subjected to a pressure of 95,000 psi for 7 min. As a result, a clear pellet, $\frac{7}{8}$ in. in diameter and about 0·05 in. in thickness, was formed. The spectra of pellets produced in this way have been found to be free from scatter losses and to contain no extraneous absorption bands.

• *STRESS RELIEF DURING MACHINING OPERATIONS*

When machining a workpiece which is rigidly supported at more than one point, stresses are set up as a result of metal removal. If it is desired to finish-machine the workpiece to a high degree of accuracy, it is necessary, before making the final finishing cuts, to loosen the clamps holding the workpiece, so as to permit relative movement of the constituent fibres of the material, thereby releasing these stresses. The clamps are then re-engaged and the machining operation is completed. During the machining of long slender workpieces, such as a turbine or compressor blade of a gas turbine, by rotation of the workpiece relative to a milling cutter or grinding wheel, it is preferable to drive the workpiece from both ends by gripping them in a pair of chucks carried by two coaxial synchronously driven work-spindles. Stresses, however, are set up in the blade and, before making the final finishing cuts, it would be advantageous to be able to free one end of the



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blade in such a manner that this end of the workpiece can twist or rotate relative to the other end and thus release such stresses. A device has now been invented in which the workpiece is clamped at opposite ends and is driven from both ends. In order to release the stresses during machining, one end of the workpiece is freed, so that the prevailing stresses can displace that end ; at the same time, however, rotation is maintained by the drive at the other end. After stress release has taken place, the clamping action is restored, so that the workpiece is again driven at both ends.

● *TRANSISTORIZED POWER-LINE CARRIER EQUIPMENT*

It is reported that the first completely transistorized power-line carrier assemblies have recently been put into operational service, the equipment being installed at each end of a 132-kV transmission line. The use of transistors in this type of equipment is claimed to present several marked advantages, resulting in lower operation and maintenance costs. In addition, installation time has been halved, owing to the compactness and simplicity of the new design. Instead of erecting a steel cabinet some 72 in. high, 26 in. wide, and 16 in. deep and then laying interconnecting wiring to the relay switchboard, all the new equipment is contained in a case only 22 in. high, 6 in. wide, and 6 in. deep, directly mounted on the switchboard, only one adjustment being required before the unit can be put into operation. Since no vacuum tubes are employed, the transistors require no periodic inspection, as they can be expected to last indefinitely. The power consumption of this transistorized equipment is given as 18 W, compared with 250 W for conventional assemblies, mainly because transistors require no filament current. Power output is approximately 1 W, but sufficient sensitivity has been provided to operate through a channel attenuation of 40 db, whereas previous equipment on the installation in question was designed for a 33-db channel.

● *HIGH-POTENTIAL MAGNESIUM GALVANIC ANODE*

The steadily increasing use of cathodic protection for the mitigation of corrosion in various fields has aroused considerable interest in new developments and applications of galvanic anodes. Great efforts have been made to improve the important properties of these anodes through the use of metals of high purity, of alloying agents, and of anode back-fill material. In recent studies, outstanding emphasis has been placed upon improvement in current efficiency, polarization characteristics, and distribution of corrosion attack at the anodes. The anode solution potential is also an important factor, because it is the algebraic difference between the cathode and anode half-cell potentials which determine the net driving potential; hence, assuming no change in cathode potential or circuit resistance, it can be seen that any increase in anode potential may be translated into increased anode-current output. Extensive research on magnesium-alloy anodes has resulted in the production of a new high-potential material which will deliver from 20 to 30% more current to a polarized cathode than the conventional alloy anodes at present employed. It is reported that these tests further indicate that this new anode exhibits marginally less current efficiency in normal soils, when compared with a 6% aluminium, 3% zinc magnesium alloy, whereas, in a saline environment, the ampere-hour recovery from the high-potential anode is equal to or greater than that obtained from the magnesium alloy. It is stated that a comparison of the microstructures of commercial magnesium and of the new alloy suggests that the superiority of the new material is due to the coating of the impurity particles with manganese, rendering these particles less harmful as contaminants.

● *NEW TYPE OF SECONDARY ELECTRIC STORAGE BATTERY*

A novel type of secondary electric storage battery is described in a recent patent specification, the claim being made that it is considerably lighter in weight and has an almost infinitely greater life than any known type of accumulator ; in addition, it is remarkable in that it can be reactivated from a remote position by electromagnetic waves. The new battery consists of a quartz cartridge containing a chemical mixture composed of elements whose nuclei become unstable as a result of bombardment by short electromagnetic waves, so that the elements become radioactive and also release electrical energy. The tube is located between the poles of an electromagnet, the electromagnetic field serving to control the alpha and beta rays emitted by the cartridge when it is in operation. At one end, the cartridge is filled with granulated copper, and at the other with granulated zinc, the copper and zinc being connected by external terminal caps at the tube ends. These external terminals are connected with a tuning circuit and two antennae, which correspond exactly with the antennae of the transmitter ; alternatively, the antennae may be connected directly with the terminals of the cartridge. The mixture in the cartridge, between the copper and the zinc, is composed of cadmium, phosphorus, and cobalt in specified proportions and may comprise a number of small cells in series. It is claimed that a quartz tube of 45 mm length and 5 mm inside diameter, containing 30 mg of the mixture, is capable of

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discharging some 100 V at 10 A for about one hour by operating the remote transmitter at preferably 300 Mcs for some 15 to 30 sec. A typical application of such a battery is in marker buoys in harbours, where it can be reactivated at given intervals from a shore transmitting station.

• *HIGH-STRENGTH ALUMINIUM-BRONZE ALLOY*

A new high-strength aluminium-bronze alloy is claimed to combine high mechanical properties with superior casting characteristics. The alloy, which is unique in that, unlike any present bronze alloy, it contains about 12% manganese, is essentially a two-phase (alpha-beta) alloy nominally composed of 75% copper, 8% aluminium, 2% nickel, and 3% iron, in addition to the manganese. The new alloy is said to surpass in its mechanical properties all commercially available conventional bronze alloys, inasmuch as it possesses higher yield strength, greater toughness (the Izod notched impact strength being 18 to 30 ft-lb), and longer fatigue life than any known alloy of its kind. It is whitish-gold in colour and melts at the low temperature of 1814°F. It can be readily forged, rolled, or extruded, and its superior casting properties are said to give it an outstanding advantage over conventional aluminium bronzes, which are generally considered difficult to cast. When making difficult castings, such as thin vanes, the alloy flows readily, while, in the case of heavy sections, its feeding and solidification characteristics ensure absolute soundness in the finished casting.

• *FINE-GRAINED BRASS FOR DEEP-DRAWING*

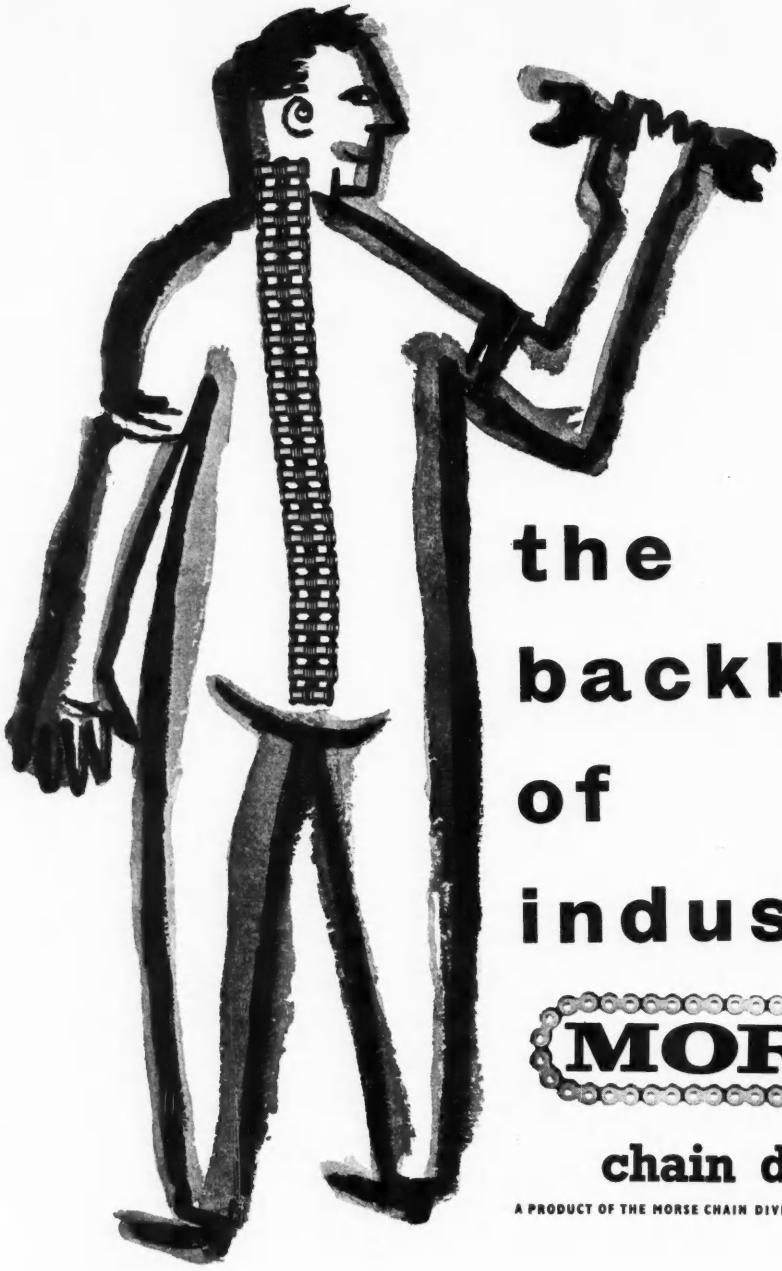
It is reported that, by means of a newly evolved continuous brass-strip annealing process, a fine-grained material is produced which is especially suitable for deep-drawing, thus contradicting the prevailing concept that a coarse grain is required for this operation. The new annealing process, which is claimed to be exceptionally fast, produces the fine-grain strip required for high finishes, in combination with superior ductility. The furnace used is of the vertical type, through which brass strip of 0·004 to 0·02 in. in thickness passes at speeds up to 190 fpm. Heating to annealing temperature is completed in less than 10 sec and is followed by immediate quenching. It is stated that the endurance limit of the continuously annealed brass for 10^6 cycles of completely reversed stress is almost double that of coarse-grained material. This is naturally of advantage in such applications as diaphragms or parts where vibration is an important factor. The continuous annealing operation requires no special atmosphere, since the combustion gases are non-oxidizing to copper and its alloys. Air and gas for firing the burners are premixed to eliminate oxygen in the products of combustion, and the furnace operates under positive pressure. Heating time is so short that zinc loss is negligible. While, in the batch-annealing process, impurities, notably iron, make close control of grain size difficult, this is not the case in the continuous annealing process, because recrystallization is rapid and at high temperature, with the iron in solution. The surface loss of zinc in batch annealing, termed scalloping, is also absent in continuous annealing in this furnace, in which the strip passes at high speed between refractory walls studded with radiant cup burners.

• *ELECTRIC TRANSDUCER SYSTEM*

A newly proposed electric transducer system is based on the fact that, if a gas-filled tube with two internal electrodes is inserted in an a.c. field of sufficient intensity to produce a discharge within the tube, a d.c. voltage will be developed between the two electrodes, the polarity of this d.c. voltage being determined by the field distribution surrounding the tube. The two electrodes may be of any shape and, in the case of a cylindrical gas tube, are, for instance, arranged near opposite ends of the tube, which is filled with an ionizable gas such as helium or neon under low pressure. A third, auxiliary electrode, preferably in the form of a ring, surrounds the tube and is held in a movable support, so that the electrode can be moved coaxially with the tube. An a.c. voltage oscillator is connected with the auxiliary electrode and one of the electrodes of the tube. Provided that the peak voltage of the a.c. potential of the oscillator is high enough to produce an electrical field of sufficient strength to ionize the gas in the tube, a d.c. potential will be produced between the two electrodes, and both the voltage and polarity of the direct current will depend upon the position of the auxiliary electrode along the axis of the tube. It is evident that this phenomenon makes the device suitable as a transducer system, an outstanding advantage, of course, being that no mechanical forces or restraints are involved. Proposed applications include its use in gramophone pick-ups, gyroscope pick-offs, strain gauges, seismographs, vibration indicators, position and angle indicators, and also electric bridges for the measurement of electrical impedances, dielectric constants, and other electrical and magnetic variations.

• *PROTECTIVE COATINGS FOR TITANIUM*

Because of its high strength-to-weight ratio, titanium ranks with steel and aluminium as a structural material. However, titanium has certain disadvantages for some applications. Thus, it tends to gall or seize when it is in loaded contact with itself or other metals, and it oxidizes



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at elevated temperatures. These disadvantages could be minimized if titanium were coated with hard, oxidation-resistant metals such as chromium or nickel, but attempts to produce such coatings have not until now been entirely successful, the main problem in depositing another metal on titanium being lack of adhesion. This has been attributed to the presence of an oxide film on the titanium, but attempts to remove it with acid etches or anodic films have not been satisfactory. In a recent investigation, several etching and electroplating procedures were tried; it is now reported that a method has been found of producing hard adherent protective coatings on titanium. The new process involves the formation of a titanium fluoride film on the surface of the metal, then electroplating it with chromium, and finally heat-treating the plated part at 800°C. It appears that the titanium fluoride film prevents the formation of an oxide and, when the specimen is placed in the plating bath, the titanium fluoride dissolves, thus permitting the chromium to bond directly to the base metal. The process is carried out by suspending the degreased and cleaned titanium part in a solution of hydrofluoric acid and acetic acid. After 10 or 15 min, an alternating current is passed through the part for another 10 min. The parts are then rinsed and transferred to a conventional plating bath, where they are chromium-plated at a temperature of 85°C.

● *EQUIPMENT FOR STUDYING ULTRASONIC PROPAGATION*

For investigations of ultrasonic signal returns as a function of frequency, a broad-band pulsed ultrasonic system is required which is capable of being frequency-modulated. The system response must be flat over the frequency band of operation and must include, not only the frequency responses of both transducers, but also the characteristics of the transmission medium between the transducers. Equipment has now been developed which can be used to study the shape of echo pulses received from objects "illuminated" by the source transducer, which consists of six cylindrical barium-titanate transducers in parallel in a single housing. The detector transducer is a single barium titanate disc transducer operating far from resonance. The broad-band response of the system is claimed to make possible accurate measurements of the time of arrival of the echo, and the flat response achieved is such that the echo received can be analysed for amplitude variation over the frequency-modulated range of 165 to 240 kcs. Resonance in the object causing the echo can change the amplitude of the received signal considerably. Another application of this equipment is in the field of seismic models, and would involve a transmission medium consisting of a number of solid layered media. Analysis of the received signal for amplitude variations as a function of frequency and/or layer thicknesses will give information concerning the interference effects of the individual reflections from the layer interfaces. It may be desirable to change the frequency of the pulsed sine-wave generator, depending upon the scale factor of the seismic model. The same principle has been used to produce pulses in the region of 25 to 150 cps.

● *ELECTROSTATIC CLUTCH*

Because of the urgent demand for increasingly higher operating speeds of the mechanical sections of electronic computers and data-processing machines, a search has been undertaken for an electromechanical transducer which would be inherently faster than the electromagnets usually employed. It was found that an electrostatic clutch, operating on the Johnsen-Rahbek effect, promised to meet this requirement, the Johnsen-Rahbek effect being the electrostatic adhesion between a metal plate and the surface of a semiconductive material under an applied voltage. As employed in clutch form, it will develop phenomenally high mechanical forces from low levels of input control energy, while high torque-to-inertia ratio and a low electrical time constant assure fast mechanical response. The clutch is non-inductive, and the operating voltages and currents are within the range of medium- and low-power thermionic valves. Although some semiconductive materials excel in some of the desired properties, exhaustive investigations have revealed the best material to be a rubber-base plastic rendered conductive by the addition of carbon. Initial tests showed that wide variations in performance, even with the same clutch, resulted from imperceptible variations in the surface condition caused by wear, heat, and abrasion. Some special type of lubrication was therefore found to be essential, in order to make the clutch a practical proposition, and both solid and liquid types of lubricant have been found to be effective. The solid lubricants include teflon and polyethylene, a block of either of these plastics, pressing lightly on the rotating surface of the clutch-facing material, providing effective lubrication. Suitable liquid lubricants include the silicone oils and certain fluorocarbon oils and waxes. Minute amounts of oil are used since, if too much is applied, torque will diminish and the residual drag at no-voltage will increase.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

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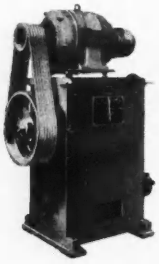

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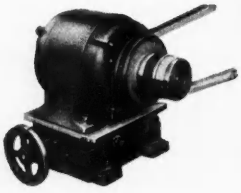

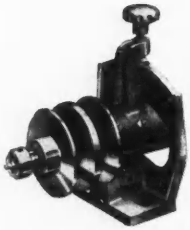

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ENGINEERS' DIGEST SURVEY No. 1

Mechanical Infinitely Variable Speed Drives

By P. CAHN-SPEYER, Dipl. Ing., A.M.I.Mech.E.

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Mechanical Infinitely Variable Speed Drives

By P. CAHN-SPEYER, Dipl. Ing., A.M.I.Mech.E.*

INNUMERABLE applications for which variable-speed drives are desirable, or even essential, have stimulated the invention of an enormous crop of devices intended to provide infinitely variable speed regulation of equipment. Of these, some have been impracticable, many have been inefficient, and others have been prohibitively complicated and costly. Nevertheless, there still remains a number of well-defined types, produced by a comparatively large number of manufacturers all over the world, which, by virtue of their efficiency and inherently sound design, have satisfactorily withstood the demands of practical requirements.

It should be understood, however, that the selection of the most suitable type of variable-speed gear for a given application is not necessarily a simple matter. Indeed, numerous aspects must be considered, if the best compromise between such factors as initial cost, efficiency, speed range, power and torque characteristics, life expectancy, maintenance, and environmental conditions is to be achieved. Besides elaborating these factors, the purpose of this article is to describe and illustrate the major types of currently available mechanical infinitely variable speed drives throughout the world, and to give some indications of the applications and particular advantages of each type, with performance data and the design features of individual makes.

Scope and Applications of Infinitely Variable Speed Drives

By definition, infinitely variable speed drives are mechanisms which convert a constant input speed into output speeds which are steplessly variable within a certain range. This range may cover a definite ratio of maximum to minimum output speed, e.g., 4:1, or it may reach from a maximum output speed right down to zero, in some cases in both directions without changing the direction of rotation of the input shaft. By their nature, then, these drives do not include stepped methods of speed adjustment providing certain predetermined fixed speed ratios, e.g., gearboxes and stepped pulleys, these constituting a class of transmission of their own, outside the scope of this article.

The important technical and economical advantages of steplessly variable speed regulation may be assumed to be generally recognized and appreciated. In some cases, the provision of variable speed in a given operation may be inherently necessary, while in others it may be at least desirable. Thus, for example, the provision of variable-speed devices for ensuring optimum production rates with variations in raw materials, for providing constant cutting speeds in machine tools in accordance with varying work diameters, for compensating effects of viscosity, temperature, and humidity in processing plants, and for synchronizing the flow of material from one machine to another may well spell the difference between profitable and uneconomical production. On the other hand, there are certain cases where a definite number of fixed speeds in steps will provide all the necessary speed variations throughout a required range, making an infinitely variable drive an unnecessary luxury.

It will therefore be appreciated that not only is it necessary to decide whether, in fact, an infinitely variable speed drive is essential or advantageous in a given application, but also, if the answer is in the affirmative, whether the device should require speed adjustment whilst in motion, or whether speed setting with the machine stationary would fulfil the purpose equally well. In this connection, there are undoubtedly many applications in which it is desired to obtain one fixed optimum speed which, however, can be established only by trial and error. In such cases, the stationary-control type of variable-speed device, which consists essentially of axially adjustable sheaves providing variable pitch diameter for a belt, furnishes an inexpensive yet fully adequate solution to the problem. Stationary-control types of variable-speed drives are, however, well known, and are not considered in this article, which is confined to those capable of adjustment during motion. Nevertheless, it is worth mentioning that, in the writer's experience, it is by no means uncommon to find installations fitted with the latter type of variable-speed drive which, after initial adjustment, has been running and is likely to continue to run at one fixed speed. Obviously, then, a little more attention to the original planning of these installations would have resulted in substantial savings in outlay and reduced maintenance costs, as the stationary-control type of drive could have been initially installed with equally satisfactory results.

Factors Influencing Type Selection

Once it has been decided that an infinitely variable speed drive is desirable in a given application, certain fundamental considerations influencing its selection must be carefully studied ; for convenience, these are analysed in the following.

(a) Required Range of Speed Variation:— It might appear, at first sight, that maximum and minimum output speeds from a speed changer are given *a priori* by the required speed ratio of the driven machine. This may indeed be true where earlier experience of any particular machine, processing plant, conveyor, etc. is sufficiently extensive to determine present and estimated future drive requirements with a reasonable degree of accuracy. Suppose, for example, that the immediately required speed ratio for a machine is 3:1, with a possible future increase to 4:1. Suppose also that the prospective purchaser is satisfied as to the suitability in all other respects of one particular type of speed changer, whose nearest standard ratios are 3:1 and 6:1. In this case, the selection of a drive with a 6:1 ratio, " to play safe ", may, from an economical point of view, prove to be quite wrong, since a drive with a markedly higher speed ratio than that required will not only involve greater capital costs, but will also probably have an appreciably reduced service life and require higher maintenance. As a general rule, therefore, selection of equipment with the lowest possible speed ratio compatible with drive requirements is recommended, particularly in the case of belt drives. At the same time, it is important to bear in mind that the

* Chief Editor, *The Engineers' Digest.*

function of a variable-speed gear should never be relegated to that of a mere speed reducer. In other words, the maximum speed of the driven machine should always correspond with the highest speed available from the variable-speed gear.

(b) Horsepower and Torque Requirements:— In the majority of cases, the driven machine requires *constant torque* over its entire speed range, i.e., the horsepower absorbed decreases in direct proportion to speed. This applies, for instance, in printing machines, conveyors, packaging machines, compressors, etc. Another group of machines, notably machine tools, requires *constant horsepower* over the whole speed range, i.e., it is necessary that the power of the drive motor be fully utilized, even at the lowest operating speed. In a third group (e.g., fans), torque will vary exponentially with speed.

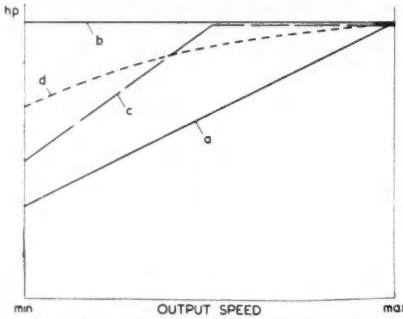


Fig. 1. Horsepower characteristics of variable-speed drives.

As will be seen later, most types of infinitely variable speed drives are rated on a constant-torque basis, as shown in principle by curve (a) in Fig. 1. Some, however, are capable of transmitting constant horsepower (curve (b)), while others are rated on a basis of constant horsepower over one part of their speed range and constant torque over the other (curve (c)). An intermediate between constant-torque and constant-horsepower rating is also possible, as indicated by the example in curve (d).

It is clear, therefore, that, for a machine requiring constant-torque input, the drive must be so dimensioned as to be capable of transmitting the maximum horsepower at the highest speed that can be developed. On the other hand, in a constant-horsepower application, a drive with constant torque must be capable of transmitting the maximum horsepower at the lowest speed; in this case, in the higher speed range, the horsepower capacity of the drive will be far in excess of the motor rating. In those cases where torque varies with some power of speed, e.g., fans, the horsepower capacity at the highest speed will be the selection criterion, inasmuch as slight overspeeding could easily result in damage to the transmission. On the other hand, power transmission capacity at low speeds will be greatly in excess of requirements.

(c) Type of Service:—Under this heading must be considered a number of factors, each of which may have a decisive influence on drive selection, and which, for convenience, have been divided into four basic groups, i.e., operational conditions, control conditions, environmental conditions, and noise considerations.

Operational conditions include such aspects as continuous or intermittent operation, frequency of starting and stopping, frequency and amplitude of load variations, frequency of shock-loading (if any), and ability to sustain possible overloads. In addition, it is necessary to consider the method of reversal, if required, i.e., whether such reversals will be effected by reversing the drive motor (in which case the variable-speed device must be capable of operating in reverse rotation), or whether reversal is to be effected by the variable-speed device itself without changing the direction of rotation of the drive motor.

Control conditions relate to the method of speed control, i.e., whether speed changes are to be effected manually, semi-automatically, or automatically.

Environmental conditions embrace such considerations as exposure to dust, fumes, humidity, corrosive atmospheres, high temperatures, etc. This category may also include such factors as susceptibility to shock and vibration from external sources.

Noise considerations may be of importance in otherwise relatively quiet locations; in this connection, it should be mentioned that, although the noise levels produced by different types of variable-speed drives vary considerably, they are very difficult to assess, and the only true criterion is by actual test.

In view of the large variety of conditions which may prevail in actual service, particularly in specialized applications, the above is naturally not a complete enumeration of the factors which should be considered under this heading.

(d) Operating and Design Characteristics:—Among other operating and design characteristics, high efficiency is, of course, desirable. None the less, its relative importance varies from one application to another and, unless a choice has to be made between two different types of drives which are in all other respects more or less equally satisfactory, efficiency must be considered in conjunction with such factors as initial cost, and maintenance and replacement costs. It must, however, be stressed that not only does efficiency vary with different types of drives but it is also influenced, sometimes to a very great extent, by the actual speed setting. In other words, the optimum efficiency given by manufacturers usually applies only to one particular speed or to a more or less limited range, so that it is desirable to examine the efficiency of a given variable-speed device at that particular speed range within which it should operate for the greatest part of its life.

In addition to such questions as the location of the drive, space and weight limitations, life expectancy, and maintenance, other operating and design characteristics influencing selection include reliability and repeatability of speed settings and permissible speed deviations (i.e., slip) with varying loads. With regard to reliability and repeatability of speed settings, there are, no doubt, applications where accuracy of speed settings is important; however, any inherent shortcomings a particular variable-speed drive may have in this respect can sometimes be overcome by the relatively inexpensive expedient of fitting a tachometer. The question of permissible limits of speed variation with varying loads is of somewhat greater importance, as some drives have an appreciably reduced output speed with increased loads, whilst others are capable of maintaining an almost constant speed throughout the load range. In some cases, this drop in output speed may actually be desirable, whereas in others it may be a definite drawback.

Another point to be considered under this heading is that of possible adverse effects on the output shaft of the drive from external sources. If the output shaft is directly connected to the driven machine, such effects are not likely to arise; however, if some intermediate transmission elements, such as belts, chains, or gearing, are interposed

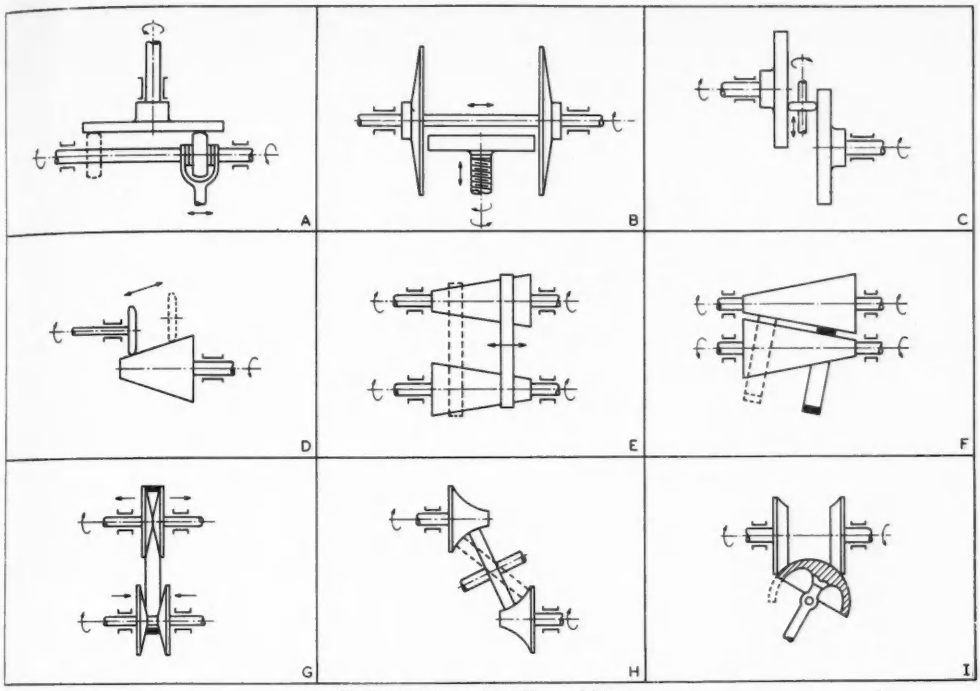


Fig. 2. Basic types of variable-speed drives.

(A) Disc and sliding-roller type, with full reversibility.
(B) Double-disc and sliding-roller type, reversible by changing contact from one disc to the other.
(C) Two discs interconnected by a sliding roller.
(D) Cone and sliding-roller type.
(E) Two cones connected by a flat, axially movable belt.
(F) Two cones connected by an endless belt.
(G) Pulleys with adjustable sheaves.
(H) Two spherical cones and free-spinning roller.
(I) Spherical cones and ball.

between the output shaft and the driven machine, external loads will be imposed on the output shaft, and care must be taken to ensure that the shaft and bearings are capable of withstanding them.

(e) **Capital Cost:**—As already mentioned in connection with efficiency, capital costs, though important, are not necessarily the overriding consideration in the selection of variable-speed drives, and must be viewed from the standpoint of overall results. As such, they must be correlated to such aspects as efficiency, maintenance, replacement, relative period of usage, and the nature of the application.

Types of Mechanical Variable-Speed Drives

Practically all modern types of mechanical variable-speed drives (as distinct from hydraulic and electrical types) can trace their ancestry to one or other of the basic types illustrated in Fig. 2, some of which have survived to the present day. Modern mechanical drives can be divided into three major types, i.e., (a) belt drives, (b) friction drives, and (c) impulse drives.

Before describing these types of drives, it should be stressed that, with *all* variable-speed drives, the horsepower capacities given in the following relate to manufacturers' rated input speeds. If lower input speeds are used to vary the output-speed range, horsepower capacities will be reduced accordingly. In addition, it must be borne in mind that the capacities given refer to ratings under normal running conditions, i.e., 6 to 10 hours operation per day, without shock loading, overloads, or frequent starting and reversals. All these ratings, therefore, are subject to a service factor or factors, depending on the relative severity of these operating conditions, and manufacturers must be consulted in each case accordingly. In the following, it is quite impossible to list the service factors for each application, as these differ considerably with different makes and types of drive. However, as a rough guide, it may be said that the service factor for shock loading may be up to 1·5, for overloading up to 1·7, for continuous duty up to 1·5, and for ambient temperature, say, 1·2 at 100°F.

A further, and very important, point is that the following descriptions are in most cases limited to the basic drives themselves, and, apart from built-in units, do not cover the possibility of combining these drives with other power-transmission elements, e.g., fixed-ratio gears, gear-shift arrangements, and differential gears. In particular, differential transmissions, in combination with almost all types of variable-speed drive, offer an extraordinary variety of output characteristics. Thus, for instance, it is possible to modify the speed range of a given drive within such wide limits that regulation down to zero and reverse can be accomplished. Moreover, with "power-bypass" systems, it is possible, at the expense of speed ratio, to transmit powers very considerably beyond the rated range of the normal drive. While combinations of this kind offer a most interesting field of study, proper coverage would require a complete article in itself, and is hence beyond the scope of this survey. However, the potentialities of combining variable-speed drives with other power-transmission elements merit investigation for unusual requirements.

BELT DRIVES

It should be understood from the outset that the majority of variable-speed requirements, particularly in the low-horsepower field, can be satisfied by the installation of one or the other variety of variable-speed belt-drive systems, the great advantages of which are their relative simplicity and low initial cost. On the other hand, all such systems have certain shortcomings which may rule them out for specialized applications. Among these shortcomings, which are present to a greater or lesser degree in practically all the drives described under this heading, is the fact that speed is not maintained strictly constant under varying and shock loads, owing to slip and belt stretch, that belts tend to wear rapidly and hence may require frequent replacement (depending on their quality), and that careful maintenance is essential.

Basically, variable-speed belt drives consist of two pulleys, connected by a belt of either standard V-section or specially wide V-section, one or both of these pulleys being adjustable to vary pitch diameter. A great variety of arrangements is possible, and these will be treated in greater detail in the following, where, for convenience, they are divided into three major systems, i.e., (1) drives embodying one variable-pitch pulley driving one fixed pulley; (2) countershaft arrangements; and (3) two variable-pitch pulleys. In all these drives, speed adjustment can be made only while the machine is running.

System 1:— In this system, one or both of the sheaves of the variable-pitch pulley (usually the drive pulley) can be moved axially, thus effecting an increase or a reduction in the pitch diameter of the belt. This axial movement can either be carried out positively, or one or both sheaves can be spring-loaded (automatically adjustable type) in such a manner as to counteract the belt forces which tend to separate them. Speed variation is obtained by changing the centre distance between the two pulleys, generally by mounting the drive motor on a sliding base. With spring-loaded sheaves, increase or reduction in pulley centre distance, and hence of output speed, is achieved merely by forward or backward movement of the motor on its sliding base. With positive sheave control, however, motor travel must be synchronized with sheave displacement.

The advantages of pulleys with only one sheave spring-loaded are that they are cheaper and that they have fewer moving parts. On the other hand, with such pulleys, belt alignment is correct for only one speed setting, unless, as is fairly common, an angled sliding base is provided for the motor, to compensate sheave displacement. Another possible means of overcoming belt misalignment is to substitute a flat-faced pulley for the fixed V-pulley (Fig. 3), thus allowing the belt to find its own alignment. Both methods of compensating belt misalignment are normally used in conjunction with a wide V-belt. If standard V-belts are installed, usually no means of compensation are provided, the penalty of increased belt wear being accepted. The closer the pulleys are together, the more pronounced will be the deleterious effects of belt misalignment.

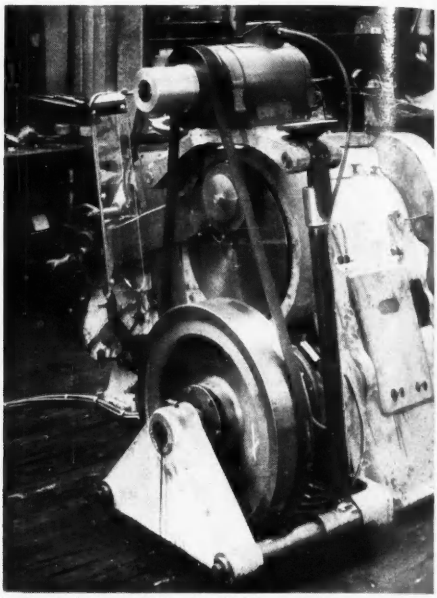
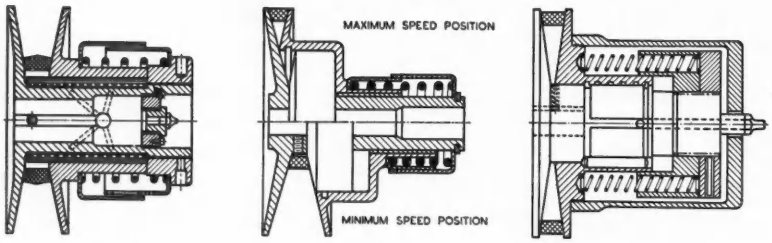


Fig. 3. Variable-speed drive installed in a printing machine, with one spring-loaded driving pulley and one flat pulley for belt alignment. (*Courtesy of Industrial Drives Limited*)

Fig. 4 shows three different designs in which only one sheave is spring-loaded. The use of one helical spring with its axis coinciding with the centre-line of the shaft (Figs. 4a and 4b) is more common than the design illustrated in Fig. 4c, in which six smaller springs are located equidistantly around the hub.

Drives with only one spring-loaded sheave are normally available for speed ratios of from 2:1 to a maximum of 4:1, with horsepowers ranging from fractional to a maximum of 15 hp at 1440 rpm input. All spring-loaded pulley drives, irrespective of whether one or both sheaves can be moved, are rated at constant torque, i.e., the ratio of maximum to minimum horsepower output is equal to the speed ratio. The horsepower capacity of spring-loaded pulley drives can be increased by fitting two sets of sheaves side by side, as shown in



((*a*) *Courtesy of Crofts (Engineers) Ltd.*)

((*b*) "*Var 'A' Cone*" *pulley by courtesy of Gerbing Manufacturing Corporation*)

((*c*) *Courtesy of Industrial Drives Limited*)

Fig. 4. Representative types of variable-speed pulleys with one sheave spring-loaded.

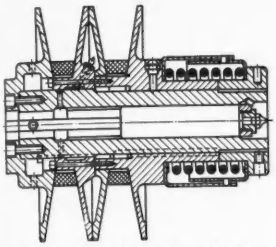


Fig. 5. **Variable-speed spring-loaded pulley for two belts.**
(*Courtesy of Crofts (Engineers) Ltd.*)

Fig. 5. However, in this case, the speed ratio is normally not more than 2:1, with an absolute maximum of 3:1.

Pulleys with *both* sheaves spring-loaded have the advantage over the type previously described in that belt misalignment is automatically eliminated. Fig. 6 shows several designs of this type, all of which use one or two helical springs, while the design shown in Fig. 7 employs leaf springs arranged radially outside the sheaves. In Figs. 6a, 6d, 6e, and 7, equal spring pressure is relied upon to produce equal sheave displacement, whereas, in Figs. 6b and 6c, equal displacement is ensured by mechanical linkage. An interesting feature of the designs shown in Figs. 6d and 6e is the use of curved faces to carry the belt, the idea being to compensate bulging of the inside of the belt when running on a small pitch diameter.

With all spring-loaded sheaves, load fluctuations will cause changes in output speed, because the elastic mounting of the sheaves tends to increase belt slip. In this connection, it is important to note that spring pressure increases with increased distance between sheaves, i.e., with decreasing effective pitch diameter, thus making speed control somewhat uncertain. It is not intended, however, to deprecate the use of spring-loaded sheaves, because in many applications accuracy of speed control is not of paramount importance; indeed, this tendency is sometimes of advantage, e.g., during starting under load and under some types of overload.

The one grave disadvantage associated with spring-loaded pulleys if they are operated at the same speed setting for prolonged periods is their tendency to "freeze" to the shaft, making subsequent attempts to change speed settings impossible without complete dismantling. This tendency is caused by what is termed fretting corrosion, which may be defined as that type of corrosion which arises when two contacting surfaces are subjected to slight relative movement of a periodic nature. It is for this reason that most manufacturers recommend that the drive should be operated throughout its full range of speed variations at regular intervals. Careful greasing is therefore very important, and attention should be paid to the accessibility of grease nipples.

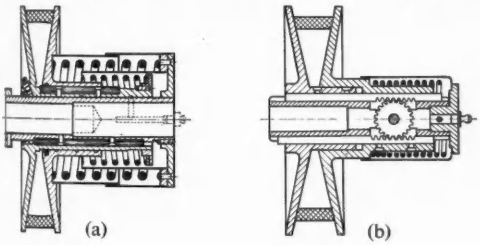


Fig. 6. **Representative designs of variable-speed pulleys with both sheaves spring-loaded.**

((a) "*Hainsworth*" *pulley by courtesy of J. H. Fenner & Co. Ltd.*)
((b) "*Roto-Cone*" *pulley by courtesy of Gerbing Manufacturing Corporation*)
((c) *Courtesy of Lewellen Manufacturing Co.*)
((d) *Standard V-belt pulley by courtesy of Lovejoy Flexible Coupling Company*)
((e) *Wide V-belt pulley by courtesy of Lovejoy Flexible Coupling Company*)

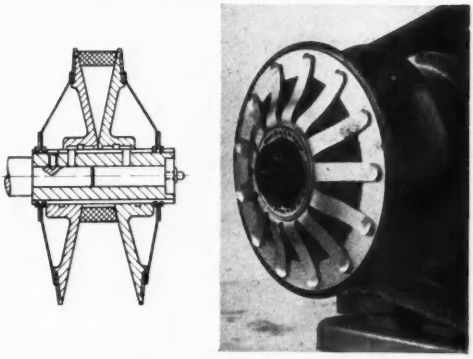
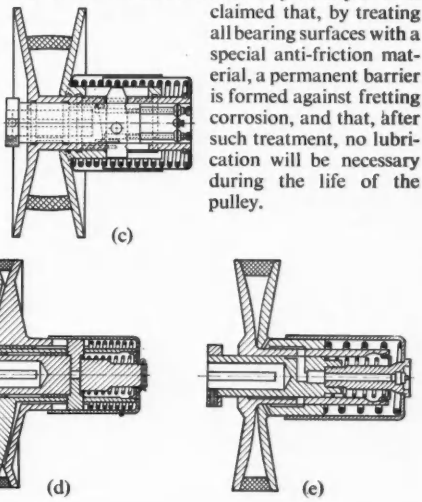


Fig. 7. **Pulley design with both sheaves spring-loaded, using leaf springs.**
(" *Simplabelt* " *by courtesy of Stahlkontor Weser G.m.b.H.*)

In an attempt to eliminate, or at least mitigate, the disadvantages of spring-loaded pulleys, a new design of pulley is being investigated in Germany by Stahlkontor Weser, in which a hollow rubber spring takes the place of the conventional helical steel spring. The idea is that this rubber spring will damp out the vibrations which may result in fretting corrosion, and achieve a more even spring pressure over the range of spring deflection. A different solution to the same problem is that provided by a design recently introduced by Industrial Drives Ltd., in which a replaceable splined ring on a female flange takes the place of the usual feather key or keys. It is claimed that, by treating all bearing surfaces with a special anti-friction material, a permanent barrier is formed against fretting corrosion, and that, after such treatment, no lubrication will be necessary during the life of the pulley.



In a third type, developed in the U.S.A. and illustrated in Fig. 8, spring-loaded sheaves are combined with six cams and followers on each face. In this design, the springs serve merely to keep the pulley faces in contact with the belt at all times, while the cam followers maintain a given speed by sliding up and down the cam tracks, in accordance with load variations. The manufacturers claim that this design assures virtually constant speed over the entire load range. Incidentally, it will be noticed from Fig. 8 that, like the design shown in Fig. 6d, this is an "interrupted-face" or intermeshing type of pulley suitable for standard V-belts.

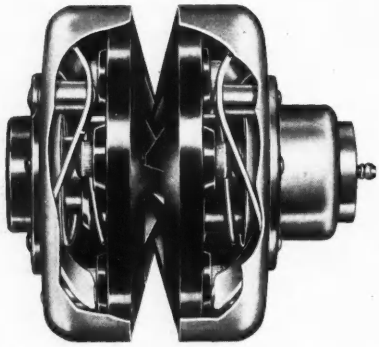


Fig. 8. Spring-loaded pulley with cam control for uniform speed under varying load.
("Hi-Lo" pulley by courtesy of Equipment Engineering Co.)

As already mentioned, pitch-changing pulleys may be of a design which effects speed changes in a *positive* manner, i.e., without the use of springs. Positive-control pulleys are somewhat more expensive than spring-loaded pulleys, but provide more accurate speed control throughout the range, in either direction, and a smooth and positive speed adjustment is conducive to increased belt life. In Fig. 9, it can be seen how sheave adjustment is effected from a screwed housing engaged by a worm, which itself is operated by a handwheel via bevel gearing and a flexible shaft. In addition, this motion is synchronized with that of the motor. Tensioning the belt, when necessary, is carried out by disconnecting the flexible shaft and repositioning the motor by means of the handwheel. A somewhat similar design, but effecting

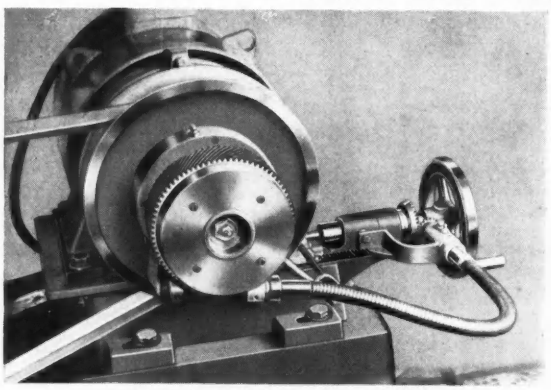


Fig. 9. Positive springless variable-speed pulley.
(Courtesy of Crofts (Engineers) Ltd.)

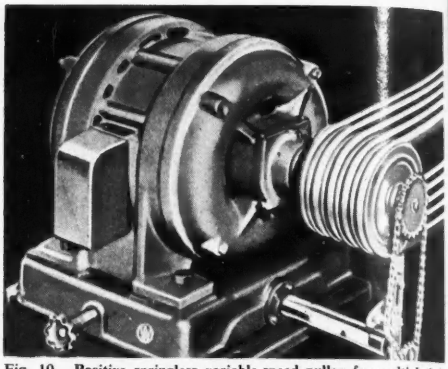


Fig. 10. Positive springless variable-speed pulley for multiple belts.
("Vari-Pitch" pulley by courtesy of Frank Wigglesworth & Co. Ltd.)

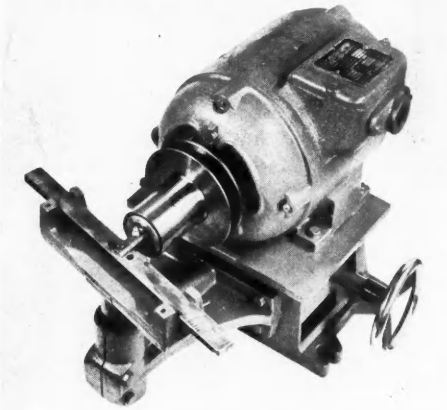


Fig. 11. Cam-operated positive springless variable-speed pulley.
("Druce-Elliot" pulley by courtesy of Industrial Drives Limited)

synchronization of displacement by chain and sprocket, and arranged for multiple belts, is shown in Fig. 10.

An unconventional method of positive control is provided by the Druce-Elliot unit illustrated in Fig. 11. In this arrangement, operation of the handwheel causes the motor to move forwards or backwards along angled slides, displacing with it the inner flange of the pulley, this flange being fixed to the motor shaft. At the same time, the outer flange of the pulley is displaced in an opposite direction, by virtue of the fact that it is guided by a slide-bar set at an angle opposite to that of the base. As a result, the pulley flanges move an equal amount in opposite directions, so that perfect belt alignment is ensured. Provision is made for compensating for belt wear and belt stretch. To ensure rotation of both sheaves, the outer and inner flanges are splined together. A somewhat similar unit is the Dean-Carter drive, supplied by Carter Bros. (Rochdale) Ltd.

In general, speed ratios and horsepower ratings for positive-control pulleys are similar to those of spring-loaded pulleys. With both types, horsepower ratings can be materially increased by the use of multiple belts, but speed ratios may be substantially reduced.

System 2:— This system, which involves the use of dual pulleys on a countershaft, is perhaps the cheapest method of providing variable speed with high speed ratios. An arrangement of this type is illustrated schematically in Fig. 12, in which a fixed-pitch motor pulley transmits power to a fixed-pitch pulley on the driven machine via

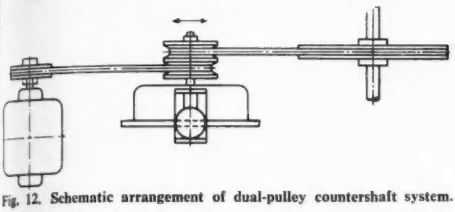


Fig. 12. Schematic arrangement of dual-pulley countershaft system.

two variable-pitch pulleys on a countershaft located between the two fixed pulleys. The central sheave of the two countershaft pulleys is floating, and speed variation is accomplished by shifting the position of the countershaft forwards or backwards, so that the effective pitch diameter of one belt increases as that of the other decreases in proportion. The relative speed range is determined by the diameter ratio of the fixed pulleys, and can be varied within wide limits. The dual pulley on the countershaft varies the speed in equal ratios above and below that speed which would obtain if the two fixed pulleys were directly connected.

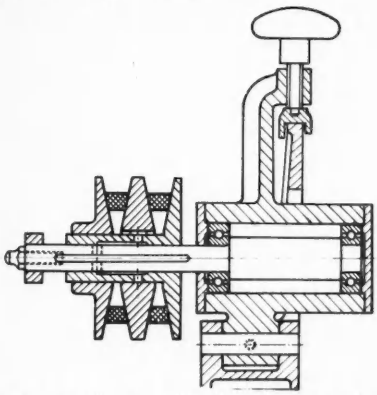


Fig. 13. Countershaft system with solid overhung dual pulleys.
(Courtesy of Crofts (Engineers) Ltd.)

Geometrical considerations show that, with this type of drive, the tension in the two belts is not necessarily constant over the speed range. If the three centres keep more or less in line, differences in belt tension will not be appreciable; in an angled drive arrangement, such differences will be more pronounced. The resulting slackness in one belt may cause slip, reducing belt life and impairing the horsepower capacity of the entire drive.

There are, however, various mechanical devices for overcoming this defect, the simplest being an arrangement in which the motor is mounted on a pivoting base. Another solution, easily adaptable to different drive configurations, is shown in Fig. 15.

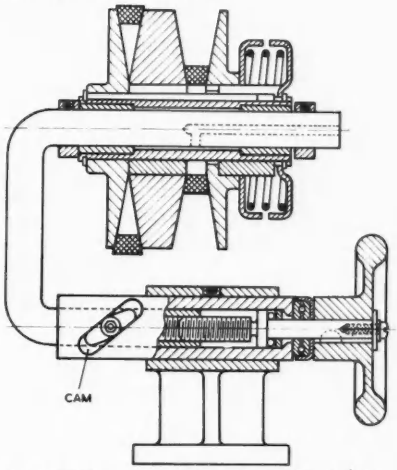


Fig. 15. Countershaft system with overhung solid pulley faces, with automatic belt tensioning and alignment.
(Courtesy of Speed Selector Inc.)

If standard V-belts are used as driving elements (generally for small horsepowers only), either solid pulley faces may be used for speed ratios usually up to 4:1 or interlocking faces for a maximum speed ratio of about 16:1. In this connection, Fig. 15 shows a design which, by virtue of a spring load on the pulley, automatically ensures equal belt tension over the whole range. This design not only adjusts the speed by swinging the countershaft in an arc around its mounting but, by means of a cam, also positively adjusts the sheave assembly along its axis to maintain belt alignment in all speed positions. However, with standard V-belts, and in view of the simplicity of such designs as those shown in Figs. 13 and 14, the penalties of belt misalignment are often accepted. Incidentally, with the arrangement shown in Fig. 14, the countershaft carries one fixed and one spring-loaded pulley. The drive motor is also provided with a spring-loaded pulley, connected with a fixed pulley on the countershaft. The spring-loaded pulley on the countershaft is connected with a fixed pulley arranged coaxially with the motor. Speed is changed by moving the countershaft towards or away from the motor and, here again, belt tension is automatically maintained through the use of spring-loaded pulleys.

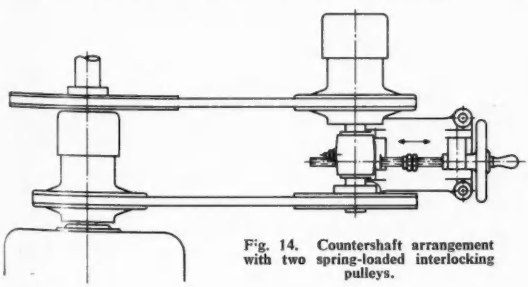


Fig. 14. Countershaft arrangement with two spring-loaded interlocking pulleys.

A different pulley arrangement, also eliminating the overhang of the arrangements shown in Figs. 13 and 15, and therefore of sturdier construction, is shown in Figs. 16a and 16b, which respectively illustrate a standard V-belt drive with interlocking faces, and a wide V-belt drive with solid faces. The example shown in Fig. 16a is mainly intended for frac-

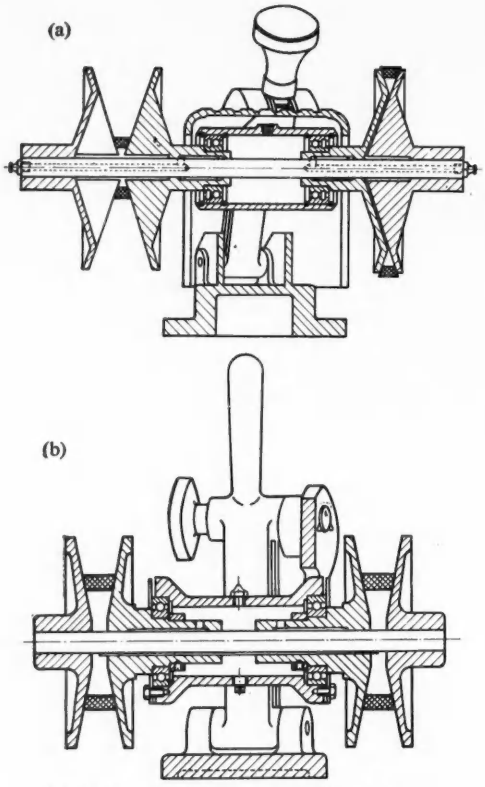


Fig. 16. Countershaft arrangement with divided pulleys.
((a) "Select-O-Speed" transmission by courtesy of Lovejoy Flexible Coupling Company)
((b) Courtesy of Crofts (Engineers) Ltd.)

tional-horsepower applications, with speed ratios up to 3·3:1, whereas that in Fig. 16b is suitable for use up to 10 hp, though it is normally used up to only 5 hp, when a 10:1 speed ratio can be achieved.

The above are only a few random examples of dual-pulley countershaft arrangements and by no means represent the full range available from any particular manufacturer. On the whole, however, all types are basically similar and vary mainly in minor constructional details and style.

System 3:—All designs in this group consist fundamentally of two belt-connected variable-pitch pulleys, and are characterized by the fact that their centre distance is fixed, or the input and output shafts remain co-axial. Several arrangements are possible, and these have here been divided into three groups, i.e., (*a*) one positively controlled pulley driving one spring-loaded pulley, or *vice versa*, (*b*) one positively controlled pulley driving another positively controlled pulley via a floating-pulley arrangement on a countershaft (tandem-belt design), and (*c*) two positively controlled pulleys directly connected by a conventional wide V-belt or by a special wide V-belt, usually wood-reinforced. Included in this group is a special, yet well-known, design in which the belt is replaced by a metallic chain.

In general, all drives belonging to System 3 have better performance and more favourable operating characteristics than those dealt with previously, but are more expensive. They are frequently built into compact, enclosed units, or as "package-type" units incorporating the drive motor and, in some cases, built-in gearing for providing the desired speed range. The fixed centre distance may often be desirable in open drives; in enclosed drives, it clearly simplifies their design.

A typical "open" arrangement belonging to *Group (a)*, giving a speed ratio of up to 8:1 with a standard V-belt, is illustrated in Fig. 17. Arrangements of this type normally have a maximum capacity of 15 hp. As can be seen, the outer sheave on the driving pulley and the inner sheave on the driven pulley are fixed, whereas the inner driving sheave is positively controlled and the outer driven sheave is spring-loaded. Belt alignment is thus maintained in all positions.

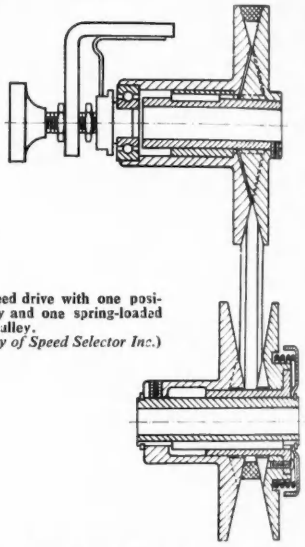


Fig. 17. Variable-speed drive with one positive-control pulley and one spring-loaded pulley.
(Courtesy of Speed Selector Inc.)

Three examples of packaged units in this group are shown in Figs. 18, 19, and 20. In all these designs, belt stretch is taken up automatically by the spring-loaded pulley.

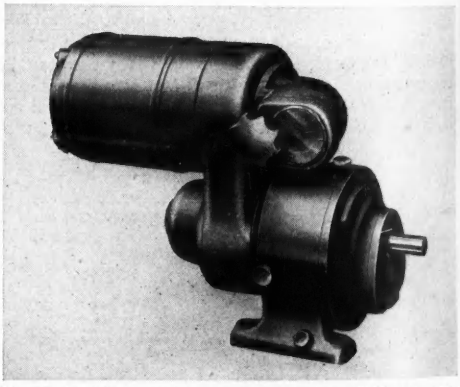


Fig. 18. Variable-speed drive with built-in motor, suitable for powers up to 2 hp and speed ratios up to 8:1.
(Courtesy of Crofts (Engineers) Ltd.)

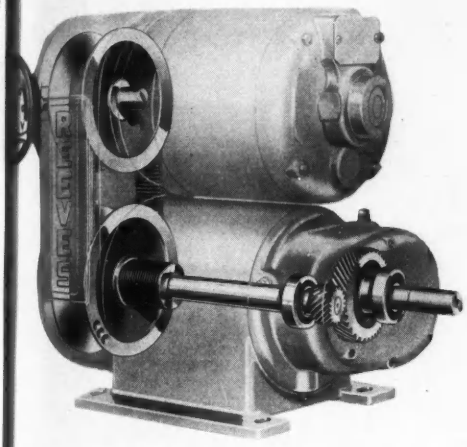


Fig. 19. **Variable-speed drive with built-in motor and reduction gear, for powers up to 40 hp and speed ratios up to 10:1.**
(*"Vari-Speed Motodrive" by courtesy of Reeves Pulley Company*)

Another drive in this group is the "Omnia" unit, manufactured in Holland by Europag C.V., in which the spring-loaded pulley is fitted to the motor, and the positively controlled pulley to the input side of a two- or three-speed gearbox. With this expedient, it is possible to obtain speed ratios of up to 25:1, with a slight overlap in each speed range selected on the gearbox.

The operation of drives belonging to *Group* (*b*) is shown diagrammatically in Fig. 21. Here, two belt drives are used, with an input shaft (1), a co-axial output shaft (2), and a countershaft (3). Each of the four V-belt pulleys consists of two cone discs, one of which is fixed, while the other is so arranged that it can slide axially. The two outer sliding cone discs on the countershaft assembly are coupled together, as are the two inner sliding cone discs on the input and output shafts, which are operated positively by a screw and handwheel (4). The cone discs on the countershaft adjust themselves automatically by virtue of the flank pressure of the belts.

Fig. 22 shows the construction of a unit built on this principle. The countershaft is equipped with a spring-loaded tensioning device, permitting belt stretch to be taken up automatically and compensating for changing arcs of belt contact. The unit illustrated is available in models up to a maximum capacity of about 12 hp, but higher capacities (up to about 60 hp) can also be supplied, in which case multiple-belt drives are fitted. The maximum speed ratio is 10:1 in the lower horsepower range, but less with higher horsepowers. Another unit operating on the same principle is shown in Fig. 23. This particular model incorporates a two-speed gearbox and a flanged motor drive up to 10 hp, and overall speed ratios between 27:1 and 35:1, depending on the horsepower. Models without gear-shift are also available, with speed ratios between 6:1 and 10:1, depending on type.

Another design belonging to *Group* (*b*) is that shown in Fig. 24. Here, as distinct from the two previous designs, the countershaft is not spring-loaded, belt tensioning being accomplished automatically by a cam-bar arrangement. As can be seen from Fig. 25, the cam bar rides on a cam roller attached to the sheave-control mechanism. The cam bar is angled at one end, so as to vary the centre distances between the countershaft and the input and output assemblies; as a result, uniform belt tension is always maintained. In addition, an auto-

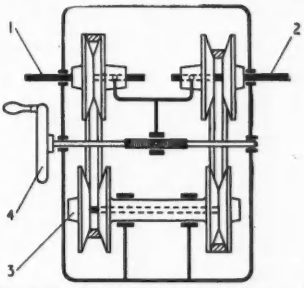


Fig. 21. **Diagrammatic arrangement of a variable-speed drive incorporating two positively controlled pulleys and floating pulleys on a countershaft.**
(*Courtesy of Eisenwerk Wülfel*

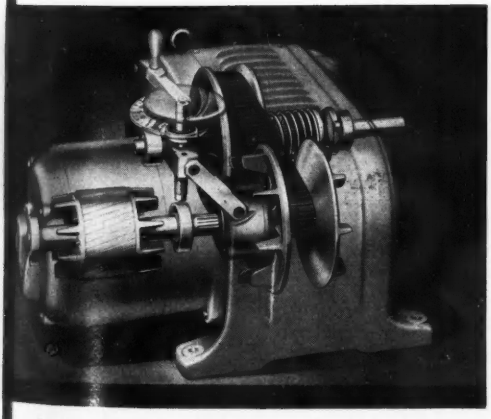


Fig. 20. **Phantom view of variable-speed drive with built-in motor, for powers up to 60 hp and speed ratios up to 10:1.**
(*"U.S. Varidrive" by courtesy of U.S. Electrical Motors Inc.*)

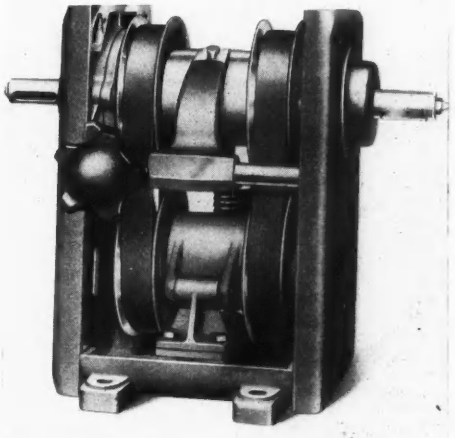


Fig. 22. **Typical variable-speed unit constructed on the principle shown in Fig. 21, with cover removed.**
(*Courtesy of Eisenwerk Wülfel*)

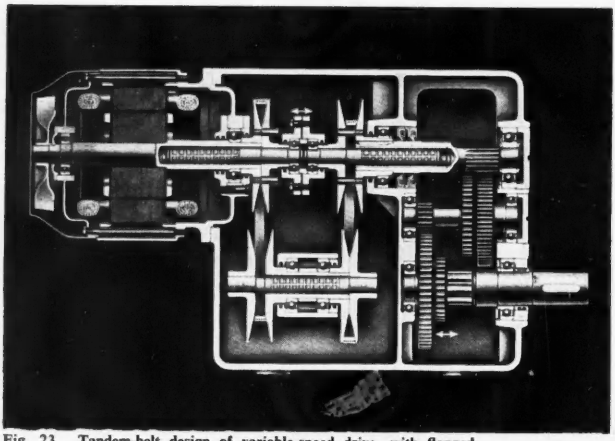


Fig. 23. **Tandem-belt design of variable-speed drive, with flanged motor drive and built-in two-speed gearbox.**
(*Courtesy of Hans Weber Maschinenfabrik*)

matic belt-tension indicator is incorporated in the design, ensuring correct adjustment of initial belt tension. This design is available in standard capacities ranging from $\frac{1}{3}$ to 5 hp, with speed ratios from 2:1 to 9:1 for the higher horsepower ranges, and up to 20:1 for the lower ranges. Special units up to 20 hp can be supplied. Recommended input speeds are between 1160 and 2000 rpm. Three speed ranges are available with each model, but the maximum output speed in all cases is higher than the input speed. These drives are rated at constant horsepower for output speeds higher than the input speed, and for constant torque for output speeds lower than the input speed.

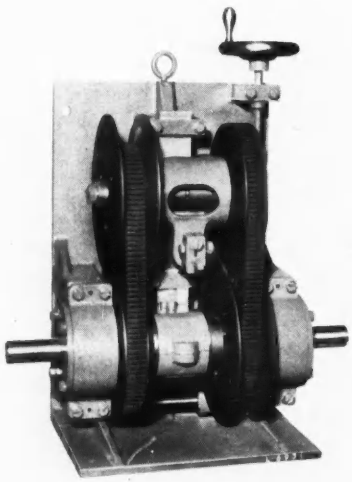


Fig. 24. **Tandem-belt design of variable-speed drive, with cam-bar arrangement for automatic belt tensioning.**
(*Courtesy of Worthington Corporation*)

A closed-type speed changer belonging to *Group* (*c*), using multiple belts and positive-control sheaves on both the driving and driven shafts, is shown in the open position in Fig. 26. In this design, sheave adjustment is

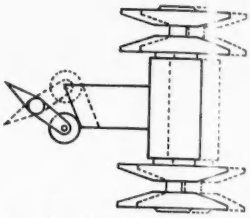


Fig. 25. **Operating principle of the cam-bar arrangement used in the design shown in Fig. 24.**
(*Courtesy of Worthington Corporation*)

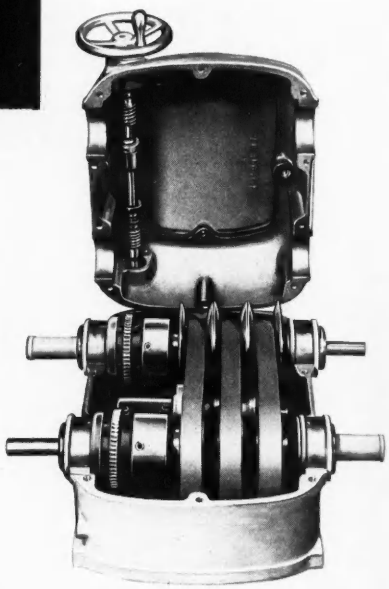


Fig. 26. **Variable-speed drive with positively controlled sheaves on both driving and driven shafts.**
(*"Vari-Pitch" speed changer by courtesy of Allis-Chalmers Manufacturing Company*)

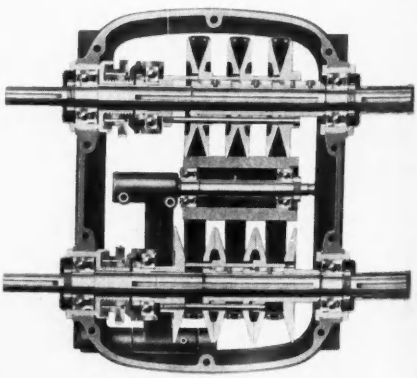


Fig. 27. **Design details of the drive shown in Fig. 26.**
(*Courtesy of Allis-Chalmers Manufacturing Company*)

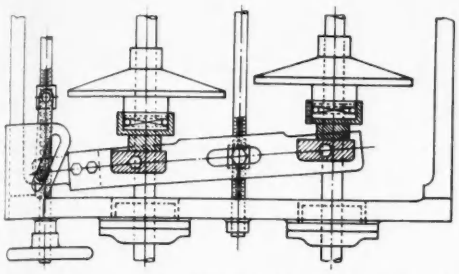


Fig. 28. **Arrangement of variable-speed drive with reinforced wide belts, showing method of speed adjustment.**
(*" Centre-Grip " gear by courtesy of Barker & Co.*)

effected in synchronism from a handwheel via two worms and wormwheels. The design features of this drive are clearly visible in Fig. 27, which also illustrates how belt tension is automatically maintained by a cantilever-type idler. These units are made in capacities from 1½ to 75 hp, with speed ratios of 3·75:1 for the lower, and 3:1 for the higher, horsepower range. The speed range depends on the input speed, which is normally limited to 1500 rpm for the low-horsepower range, and to about 800 rpm for the high-horsepower range, the lowest input speeds being in the neighbourhood of 200 rpm.

The " Motovaribloc ", manufactured in France by CIDMA Anciens Établissements Galand S.A., is another example of a variable-speed unit in which both pulleys are positively controlled. In this design, a maximum of 12 hp can be transmitted with one wide V-belt, the normal speed ratio being 4:1. These units are normally supplied complete with reduction gears, but models with two-speed gearboxes are also available.

Despite their relatively high cost and bulk, drives employing special reinforced wide belts, in combination with positive-control pulleys, have for a long time enjoyed great popularity, owing to their robustness and reliability, their high accuracy of speed control, and their

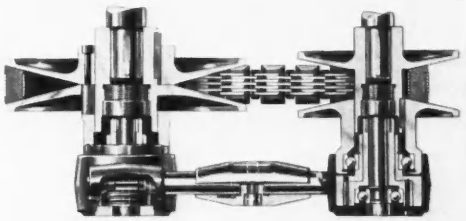


Fig. 29. **Variable-speed gear, with cones actuated and synchronized by tie-rods.**
(*Courtesy of Borgna Costruzioni Meccaniche*)

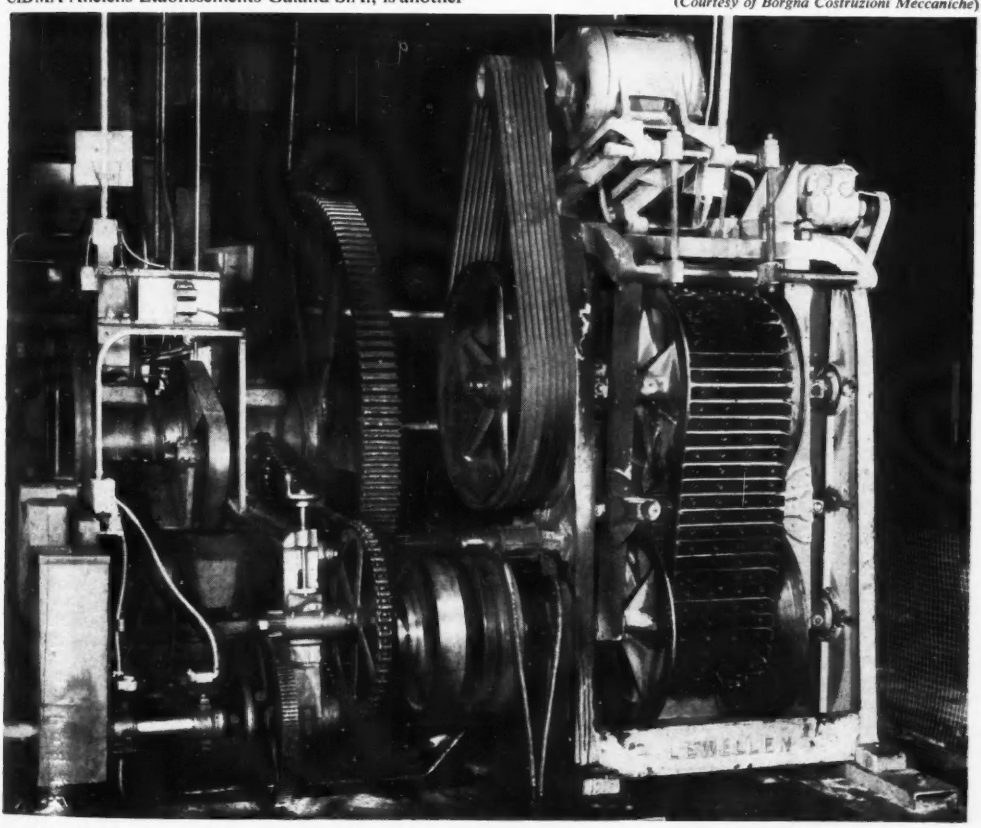


Fig. 30. **Open-type vertical transmission used for driving a veneer lathe.** (*Courtesy of Lewellen Manufacturing Company*)

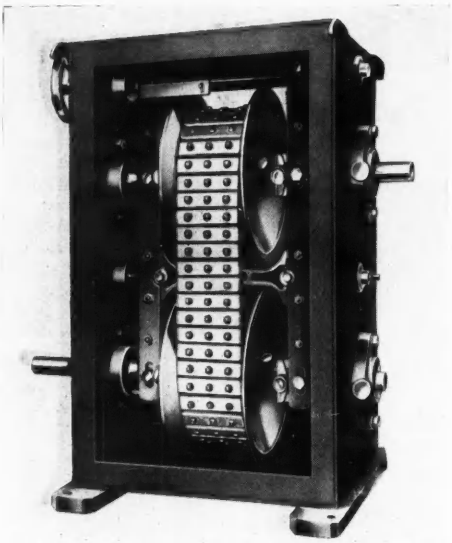


Fig. 31. Closed-type vertical transmission with cover removed.
(*Courtesy of Crofts (Engineers) Ltd.*)

high efficiency. Drives of this type are well established and are available from a number of manufacturers ; their designs, however, differ only in detail. Cone discs are shifted axially by side levers, actuated by screws and handwheels in such a way that equal belt tension is maintained at all speed settings. A typical arrangement is shown in Fig. 28, in which the central screw provides a means of adjusting belt tension. There are, however, other methods of adjusting speed and belt tension, an interesting example being a drive in which a gear mechanism operating the cones of the two pulleys is actuated and synchronized by tie-rods, as shown in Fig. 29.

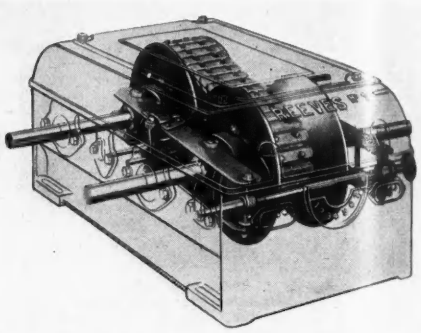


Fig. 32. Phantom view of closed-type horizontal variable-speed transmission.
(*Courtesy of Reeves Pulley Company*)

There are several styles of reinforced belts, all of which have wooden or sometimes aluminium crosspieces, bolted to the belt proper at short intervals. These crosspieces are leather-tipped and are tapered at their ends in such a way as to conform with the profile of the cone discs. The resulting wedge action prevents belt slip.

Transmissions of this type are available from most manufacturers as horizontal or vertical models, open or fully enclosed. Some representative examples are shown in Figs. 30, 31, and 32. Horsepowers range from fractional to 130 hp, though a maximum of about 80 hp is more usual. Speed ratios are normally between 2:1 and 16:1, and occasionally even 20:1, but the higher ratios are available only in the medium-horsepower range. Since input speeds are low (usually between 200 and 600 rpm) owing to the weight of the belt, speed reduction on the input side via belts, chains, or gears is necessary. Output speed range is again determined by the input speed and the speed ratio, maximum and minimum output speeds being obtained by multiplying or dividing the input speed by the square-root of the speed ratio.

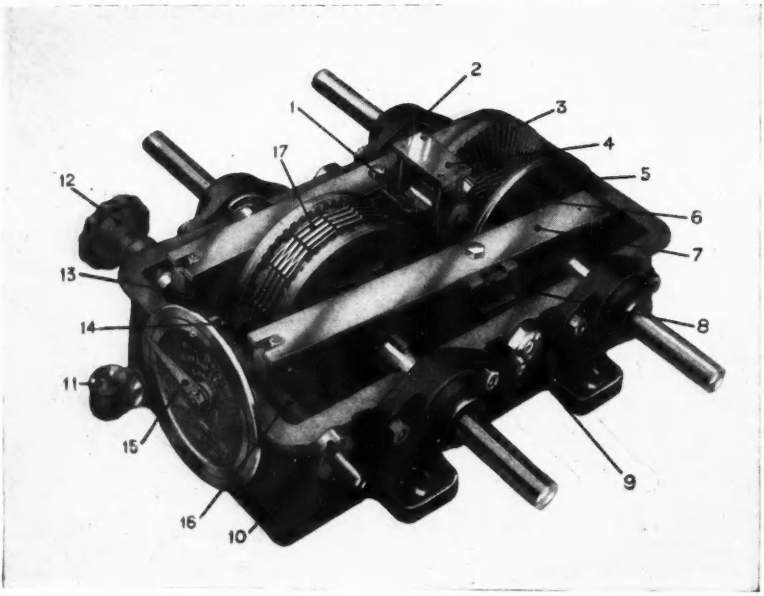


Fig. 33. Horizontal-type P.I.V. gear with top cover removed.
(*Courtesy of Stone-Wallwork Ltd.*)

(1) Shoe assembly.
(2) Radial ball bearings.
(3) Wheel face.
(4) Shoe guide.
(5) Wheel flange.
(6) Ball thrust bearing.
(7) Operating levers.
(8) Chain-tightening cam.
(9) Chain-tightening screw and locking plate.
(10) Alignment fitting.
(11) Oil filler and level indicator.
(12) Speed-control handwheel.
(13) Control bracket.
(14) Speed-limiting nut.
(15) Speed indicator.
(16) Control screw.
(17) P.I.V. slat-type chain.

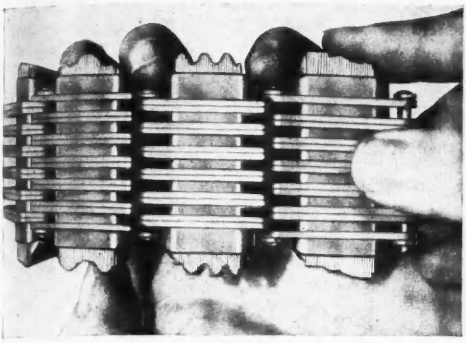


Fig. 34. **Detail of P.I.V. slat-type chain.**
(*Courtesy of Link-Belt Company*)

This square-root law also applies to estimation of the horsepower capacity at low speed, which is therefore better than that obtainable with constant-torque drives. As an example, with an input speed of 300 rpm and a speed ratio of 16:1, the maximum output speed will be 1200 rpm and the minimum will be 75 rpm; if the maximum rating of the drive at 1200 rpm is 10 hp, its rating at 75 rpm will be 2½ hp. This type of drive is deservedly popular and is marketed by numerous other manufacturers, e.g., Williams & Womersley Ltd., and Frank Wigglesworth & Co. Ltd., in England.

Another type of transmission in this general classification uses special chains, in place of belts. These drives, invented in the early 1920's, are now known universally under the name of P.I.V. (positive infinitely variable) gears. They are unique in that power transmission no longer relies on friction, but is effected in a positive manner, similar to that of a chain and sprocket. As in the case of drives with reinforced wide belts, speed changes are brought about by an axial movement of the wheels on their shafts (Fig. 33), by means of two pairs of operating levers (7) and left- and right-hand screws. Each link of the power-transmitting chain (Fig. 34) contains a cage fitted with steel slats ; these slats are able to move transversely, as well as to adjust themselves radially to fit into the radial grooves of the conical wheels (3) in Fig. 33. These wheels, which have teeth cut at a constant depth but of increasing width towards the periphery, are so mounted on their shafts that the teeth of one wheel are opposite the grooves between teeth of the other wheel. As the chain enters, the teeth of one wheel push the requisite number of slats into the spaces on the opposite wheel, so that new teeth are continuously formed. Positive chain engagement is thus ensured. The slats do not have to slide over each other when packed together under the working load, correct tooth formation being obtained, even with a slack chain, by lightly spring-loaded shoes (1).

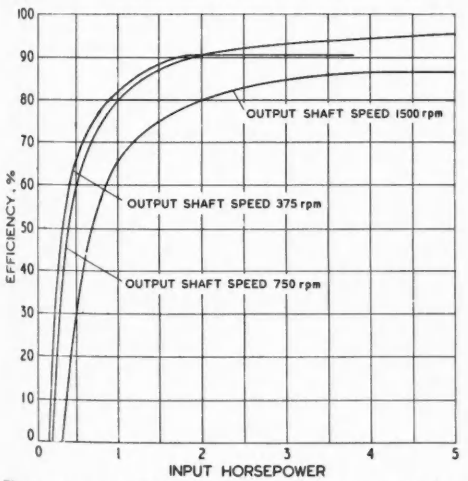


Fig. 35. **Curves of efficiency as a function of input horsepower for a P.I.V. gear with an input shaft speed of 750 rpm, and various output shaft speeds.**
(*Courtesy of Stone-Wallwork Ltd.*)

P.I.V. gears are manufactured with capacities ranging up to a maximum of 25 hp, and with a maximum speed ratio of 6:1 at medium, and 4:1 at high, horsepower ratings. Recommended input speeds vary from 500 to 950 rpm approximately, depending on the size of the unit and the specifications of different manufacturers. As before, horsepower varies according to a square-root law. Efficiencies are very high and fairly uniform over the speed and load range, as can be seen from the curves in Fig. 35. P.I.V. gears are also available with built-in differential gear units.

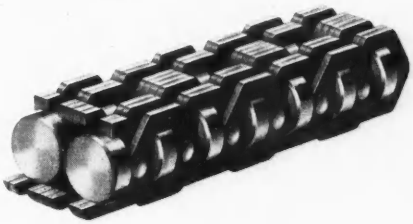


Fig. 36. **Detail of roller chain.**
(*Courtesy of P.I.V. Antrieb Werner Reimers K.G.*)

Fig. 37. **Rotary motion of rollers during (left) wedging, and (right unwedging.**
(*Courtesy of P.I.V. Antrieb Werner Reimers K.G.*)

A recent addition to P.I.V. gears is the roller-chain drive, in which power transmission is brought about by static friction between a special wedge-type roller chain and smooth conical wheels. Each link of the chain (Fig. 36) contains cylindrical rollers which, in the direction of travel, bear against the enclosing cage-like side-bars of the links, but which are free to rotate about their axes. When the chain enters the wheels, the rollers of each link wedge themselves tightly into the vees formed by the conical wheels, carrying out a rotary motion on the working surfaces of the wheels and on their own lines of contact. This wedging and unwedging action is shown in Fig. 37. The capacities of roller-chain gears are appreciably lower than those of slat-type chains, but higher speed ratios are possible. Thus, 10:1 speed ratios are available from fractional to a maximum of 4 hp at input speeds of 920 rpm, dropping to 4:1 at 6½ hp at 1400 rpm.

FRICTION DRIVES

The distinguishing feature of friction drives is that rotary motion is usually transmitted from one metallic part to another metallic part by friction generated through point or line contact. Speed changes are then made possible by altering the position of the point or line of contact relative to the centres of rotation of the driving and driven members. In their simplest form, such drives are illustrated in Figs. 2A, B, C, D, H, and I.

To-day, friction drives have attained a high degree of perfection and have been developed into a number of well-established designs, which have given excellent service in the most varied applications. The crucial problem confronting the designers of such drives is that, since transmissible power depends on the magnitude of frictional drag, and friction in turn depends on the force applied at the point of contact, specific contact pressure must be very high ; consequently, wear is considerable, even if, as is actually the case, the transmission parts are running in an oil bath. Although this problem has to a great extent been overcome by the provision, in one form or another, of more than one contact point or by line contact, the fact remains that, with only a few exceptions, friction drives are strictly limited in their horsepower capacities, generally not exceeding about 15 hp.

One variety of friction drive exists, which may be called " part-metallic ", in which one of the contact members is made of a non-metallic friction material. Whilst part-metallic friction drives are not as numerous as " all-metal " types, they have been very successfully employed, particularly on the Continent. Since part-metallic friction drives may be considered the forerunners of all-metal friction drives, they will be described first in the following.

Part-Metallic Friction Drives

Of part-metallic friction drives, probably the two best known are the EL drive and the Prym drive, both developed in Germany, where they are used extensively in machine tools. In the EL drive (Fig. 38), the motor and cast-iron driving disc (a) are held in a slide (b) and can be moved up and down by a handwheel (c) via a spindle (d), to which a speed dial (k) is geared. The driven disc carries an easily replaceable ring (e), made of a friction material. Movement up or down of disc (a) will change its radius of contact with ring (e), thus altering the speed of the output shaft (h), which is coupled to the driven disc through helical gears (f) and (g). Disc (a) and ring (e) are pressed together by a spring (i), as well as by the torque-dependent axial force exerted by the gear teeth.

EL drives are manufactured in capacities from about 1½ to 6 hp, with speed ratios of 3:1 approximately, i.e., with a normal motor speed of 1420 rpm, output speeds range from about 300 to 1000 rpm. The drive may, if required, be provided with pole-changing motors, so as to double the speed ratio. Since the main application of these units is in machine tools, which, as pointed out earlier in this article, usually require constant horsepower output over the whole speed range, the capacities given above refer to motor ratings. The operating parts of the transmission are so dimensioned that not only can full horsepower be transmitted at the lowest output speed, but overloads (corresponding to the maximum overload to which the motor can be subjected) can be sustained for short periods. The elasticity of the drive permits the absorption of shock loads, and rotation in either direction is possible. Furthermore, while EL drives are normally supplied as self-contained units, they lend themselves to incorporation into various machine tools.

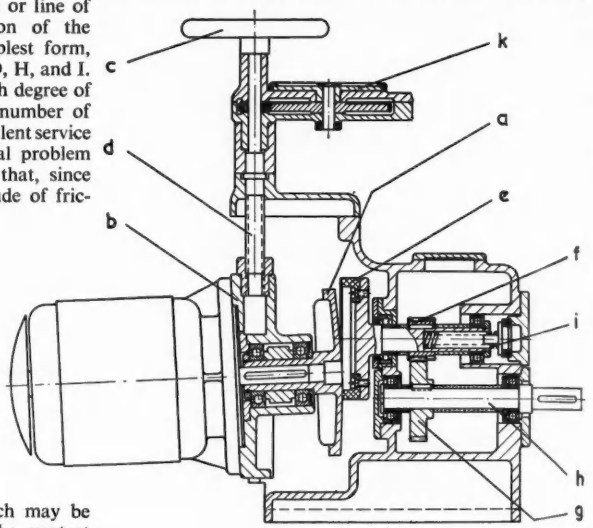


Fig. 38. **Cross-section of EL drive.**
(Courtesy of Rich. Hofheinz & Co.)

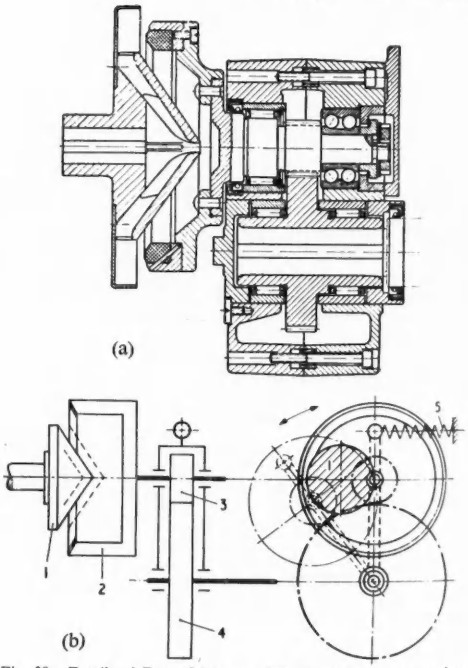


Fig. 39. **Details of Prym drive, type PK, showing (a) cross-section, and (b) method of operation.**
(Courtesy of The Motor Gear & Engineering Co. Ltd.)

Another drive of this type, similar in some respects to the EL drive, is the Stöber variable-speed unit, which has applications in the textile, as well as in the machine-tool, industry. The Stöber drive is available in a variety of models, with ratings up to a maximum of 5 hp and speed ratios between 3:1 and 5:1.

Prym drives are manufactured in three different designs, to suit various requirements of horsepower and speed ratio. Of these, the earliest is type PK, illustrated in Fig. 39. Depending on its size and input speed (1500 or 3000 rpm), this drive is capable of delivering torques from about 1·2 to 166 lb-ft, with a speed ratio of 5:1.

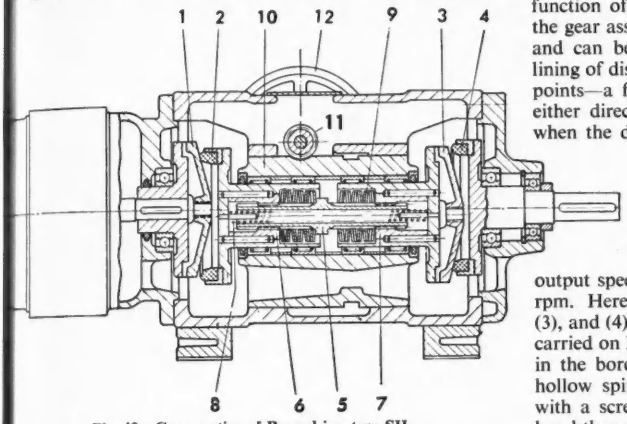


Fig. 40. **Cross-section of Prym drive, type SH.**
(*Courtesy of The Motor Gear & Engineering Co. Ltd.*)

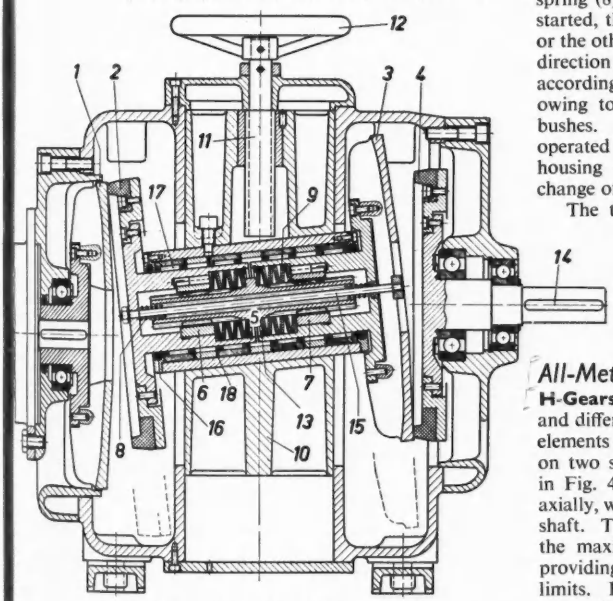


Fig. 41. **Cross-section of Prym drive, type SHK 4.**
(*Courtesy of William Prym G.m.b.H.*)

(1) and (3) Driving discs. (2) and (4) Driven discs. (5) Spindle with left- and right-hand threads. (6) Nut with right-hand thread. (7) Nut with left-hand thread. (8) Spring. (9) Belleville washers. (10) Cylindrical slide. (11) Control spindle. (12) Handwheel. (13) Circlip. (14) Output shaft. (15) Tie-rod. (16) Felt seal. (17) Roller bearing. (18) Distance piece.

The operation of type PK will be readily understood by reference to Fig 39b, in which a driving cone (1), movable axially, engages with a cup-shaped disc (2), the rim of which is provided with a friction lining. Disc (2) is connected to a planetary gear (3), which meshes with a gear (4) on the output shaft. The contact pressure between driving cone (1) and disc (2) is produced by tooth reaction forces of the planetary gear (3), and is thus a function of the torque transmitted. The body housing the gear assembly can be pivoted about the output shaft and can be swung to either side, so that the friction lining of disc (2) can contact the driving cone at opposite points—a feature enabling the drive to be operated in either direction. The weight of the housing itself (or, when the drive is in the position shown in Fig. 39b, a tension spring (5)) provides initial contact pressure.

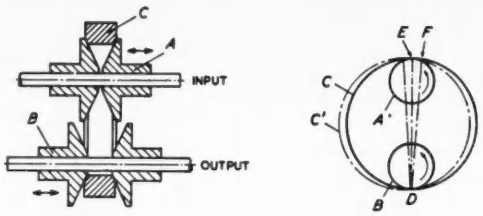
Type SH, made in four sizes, with motor capacities ranging from ½ to 4 hp, is illustrated in Fig. 40. With this design, speed ratios of 10:1 are achieved, maximum output speed corresponding to the motor speed of 1400 rpm. Here, two pairs of driving and driven discs (1), (2), (3), and (4) are arranged in series, parts (2) and (3) being carried on hollow spindles, which are free to float axially in the bore of an eccentric housing (10). Inside these hollow spindles are bushes (6) and (7), which engage with a screw spindle (5) provided with left- and right-hand threads. Initial contact pressure is established by a spring (8) carried inside spindle (5). When the drive is started, this spindle will travel until it abuts against one or the other set of Belleville springs (9), depending on the direction of rotation, and will increase contact pressure, according to the torque applied to the output shaft, owing to the action between the screw and threaded bushes. To vary the speed of the drive, a worm (11), operated by a handwheel (12), rotates the eccentric housing (10) into a different position, resulting in a change of the effective contact radii.

The third design, the latest in the Prym range, is type SHK 4, in which the capacity has been stepped up to 10 hp, with a speed ratio of 6:1. This design is illustrated in Fig. 41, from which it will be seen that, although type SHK 4 is in many ways similar to type SH, a different speed-change arrangement is used

All-Metal Friction Drives

H-Gears:—Available from several manufacturers, and differing only in detail design, the basic transmission elements of H-gears consist of a floating steel ring rolling on two steel double cones, as shown diagrammatically in Fig. 42a. One cone of each pair can be displaced axially, whilst the other is rigidly attached to its respective shaft. Thus, the ring can occupy any position between the maximum and minimum diameters of the cones, providing infinitely variable output speeds between these limits. Line-contact pressure between the ring and the cones develops automatically and is proportional to the load. In Fig. 42b, driving cones *A*, driven cones *B*, and ring *C* all rotate in an anticlockwise direction. Load application causes the position of the ring to change from *C* to *C'*, transferring the contact from *E* to *F*. Since distance *DF* is shorter than distance *DE*, the resulting deformation of the ring will increase contact pressure.

A typical example of an H-gear is illustrated in Fig. 43. Capacities of H-gears range from fractional to about



(a) (b)

Fig. 42. Basic power-transmission elements of the H-gear.
(*Courtesy of E. H. Jones Machine Tools Ltd.*)

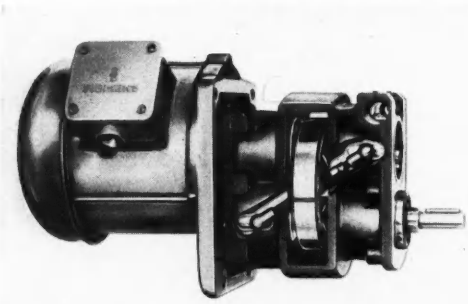


Fig. 43. Cut-away view of an H-gear.
(*Courtesy of Hans Heynau*)

5 hp, according to the manufacturer. Speed ratios are normally 6:1, 8:1, or 9:1. However, one American design, made by The Master Electric Co., is claimed to have a speed ratio as high as 15:1 with the smaller units.

These drives are available with or without built-in motors, and reduction gears can be incorporated, to give the desired range of speed. Whilst it is claimed by the makers that slip is practically eliminated under normal load conditions, H-gears are sensitive to shock and overload, damage to working parts under such conditions being a possibility. However, a safety clutch on the input side will afford effective protection.

The Graham Variable-Speed Transmission:— These drives, very well known in the U.S.A., are particularly attractive when a wide speed range is required and horsepower demands are moderate. The operating principle of these drives is similar to that of a compound planetary-gear system, except that the non-rotating member is a traction ring, which engages tapered rollers at varying diameters. As can be seen from Fig. 44, the traction or control ring can be moved axially in order to change speed. The rollers are supported in a carrier, and are inclined at an angle equal to their taper, so that their outer edges are maintained parallel to the central axis. The required traction pressure between rollers and ring is obtained through the centrifugal force of the rollers themselves, which press outwards against the ring. The output shaft derives its speed from the planet pinions. Because they move with the carrier, rotate at constant motor speed, and roll within the control ring, these pinions must at the same time turn about their own axes in the opposite direction. As the control ring is moved towards the small end of the rollers, the latter, with their pinions, turn faster in the ratio of the relative diameters of the rollers and the bore of the ring, until the speed of the pinions about their own axes equals their speed about the central axis, at which point the output speed becomes zero. Beyond this point, the output shaft reverses. The phantom view in Fig. 45 clearly shows the working parts of this transmission.

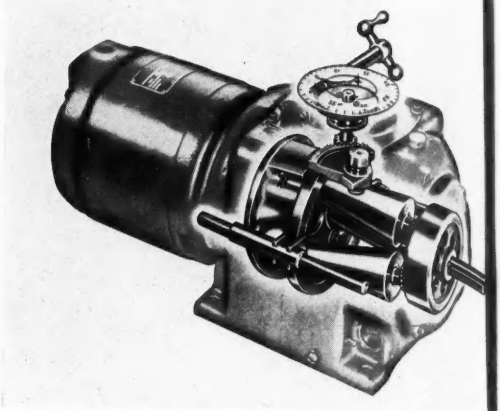


Fig. 45. Phantom view of the Graham transmission.
(*Courtesy of Graham Transmissions, Inc.*)

The Graham transmission is the first example in this survey of a variable-speed drive which can be controlled down to zero output speed and which, by suitable selection of the number of teeth in the pinions and ring gear, can reverse rotation of the output shaft, without changing the direction of rotation of the input shaft. With this type of drive, the speed ratio becomes infinity and is therefore useless as a characteristic parameter.

Standard models of the Graham transmission are available in capacities from fractional to 3 hp, with or

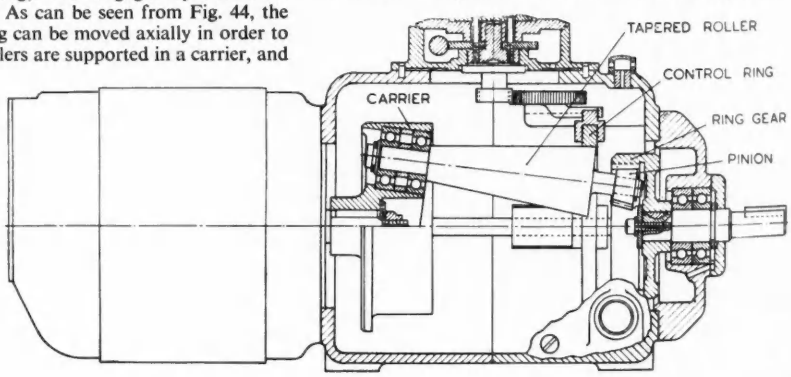


Fig. 44. Design features of the Graham variable-speed transmission.
(*Courtesy of Graham Transmissions, Inc.*)

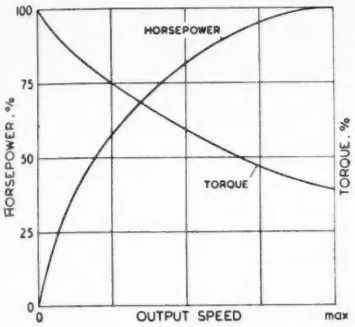


Fig. 46. Typical horsepower and torque characteristics of the Graham transmission.
(Courtesy of Graham Transmissions, Inc.)

without built-in motors, and with speeds down to zero and moderate reverse, or with equal speeds on both sides of zero. In the first type, output-speed range is from approximately 33% input speed forward to 1% input speed reverse; in the second type, the output-speed range is approximately 20% input speed in either direction. Naturally, however, the output range can be reduced or increased by a suitable built-in gearbox. In this connection, it is important that the maximum speed of the driven machine corresponds to that of the variable-speed unit. This is necessary to develop the full rated power and to distribute the speed range over the full operating length of the rollers, so as to achieve maximum accuracy and sensitivity of speed control. The input speed is normally 1800 rpm.

An outstanding feature is the protection against excessive torque afforded to the driven machine, without damage to the transmission itself. Typical horsepower and torque characteristics of the Graham transmission are given in Fig. 46. Maximum efficiency is 85% at full load for units between $\frac{1}{6}$ and 3 hp, and 65% for units of less than $\frac{1}{6}$ hp.

The F.U. Variable-Speed Unit:—The mechanism of the F.U. variable-speed unit is a development of the basic drive shown in Fig. 2C, with the sliding roller replaced by a double-conical roller. These units are made with one roller for capacities up to about $3\frac{1}{2}$ hp (Fig. 47a), and with two rollers (Fig. 47b) for higher capacities.

Speed adjustment is by means of a screw which changes the position of the roller (or rollers) relative to the driving and driven plates against which they are pressed. In the two-roller design, there are two input plates, and these are gear-driven from the input shaft, which is in this case coaxial with the output shaft. Power transmission is by line contact. The working parts of a typical two-roller design are clearly shown in Fig. 48

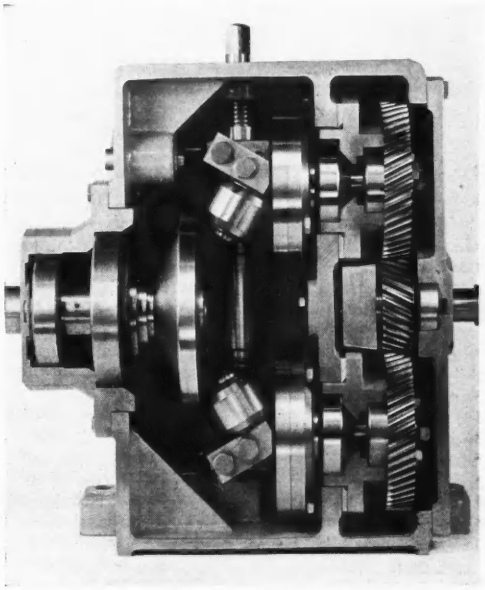


Fig. 48. Cut-away view of typical F.U. two-roller unit.
(Courtesy of F.U. (London) Ltd.)

Geometrical considerations show that true rolling can take place in only one position, i.e., at the mean speed setting. Movement of the roller away from this position gives rise to a small departure from ideal, true rolling conditions, but the ensuing skidding, due to the difference in peripheral speed across the narrow contact face of the roller, is very slight. The effect of this skidding is, however, reflected in the efficiency curve (Fig. 49),

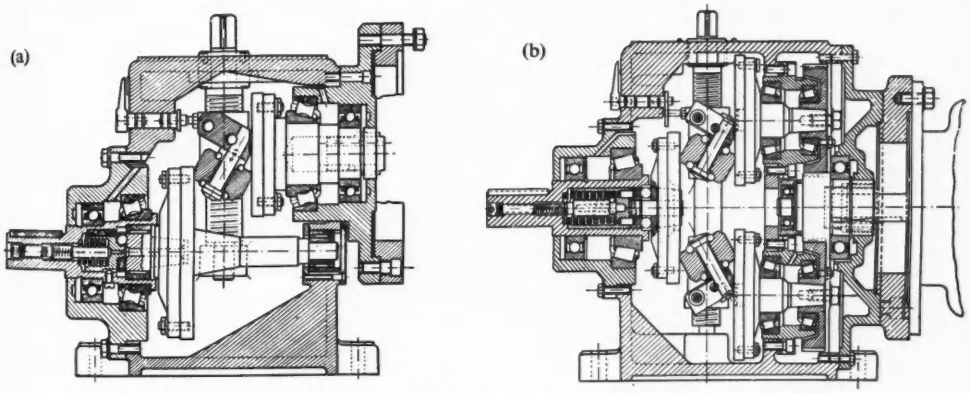


Fig. 47. Cross-section through the F.U. variable-speed unit, showing (a) a one-roller design, and (b) a two-roller design.
(Courtesy of F.U. (London) Ltd.)

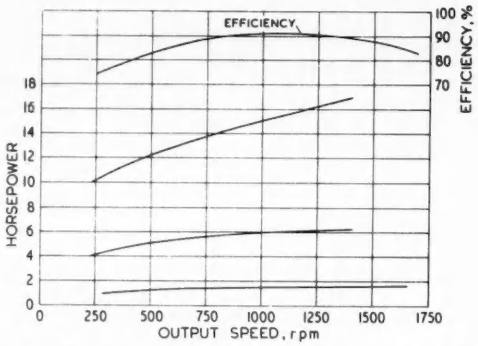


Fig. 49. Horsepower characteristics and efficiency curve of the F.U. variable-speed unit, with an input speed of 1430 rpm.
(Courtesy of F.U. (London) Ltd.)

which, for a two-roller design, indicates a maximum efficiency of over 90% at approximately the mean speed position. The corresponding values for a single-roller design are approximately 10% lower.

F.U. units are available in several sizes, from fractional to 17 hp. Speed ratios are normally 6:1, and the output-speed range, with an input speed of 1430 rpm, is approximately as shown in Fig. 49. The larger units can also be supplied with speed ratios of 10:1. Horsepower variations over the speed range are also plotted in Fig. 49 for the largest, a medium-size, and the smallest unit. The comparatively small losses in horsepower capacity at low speeds are attributable to a special automatic pressure device fitted on the output shaft. This device varies the contact pressure between rollers and plates in direct proportion to the load. Speed holding of these units is excellent, and it is claimed that speed does not vary by more than 0·2% over the entire load range.

The Arter Variable-Speed Gear:—As in the F.U. unit described above, rollers are the basic speed-changing elements in the Arter gear, but, as can be seen from Fig. 50, their arrangement is totally different. The conoidal wheel (2) is driven from the input shaft (1) through a torque-responsive mechanism (9) and (10). Two opposite rollers (5) are pressed against wheel (2), as well as against the driven wheel (3), which is keyed to the output shaft (4). The rollers (5) can be swung about pivots (6) to adjust the required output speed. These two pivots (6) are carried in a floating frame (7), so that the reaction forces of the rollers are balanced and are not transmitted to the outer casing. The rollers (5) rest on the cambered heads of bolts (8), permitting equal load

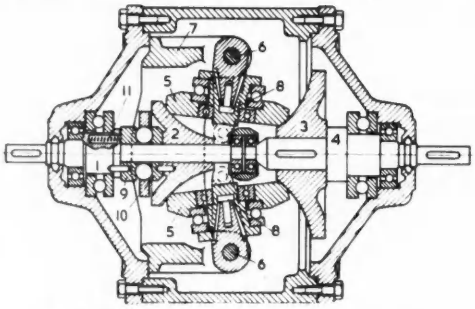


Fig. 50. Sectional view of the Arter variable-speed gear.
(Courtesy of J. Arter & Co.)

distribution of wheels (2) and (3). A spring (11) maintains minimum pressure on wheel (2) for starting and no-load operation, when the torque-response mechanism (9) and (10) is inoperative. Fig. 51 shows the working parts of the Arter variable-speed drive.

The capacity of this Swiss-made gear ranges from 0·5 to 10 hp, with a 1500-rpm input ; the speed range is from 1·6 to 0·22 times the input speed, giving a ratio of about 7:1. Smaller models (¼ hp at 1400 rpm) are also available, and for these the speed ratio is increased to 10:1. The efficiency is better than 90% over the whole speed range, and the rating is for constant horsepower with output speeds above 1000 rpm approximately, and for constant torque at lower speeds.

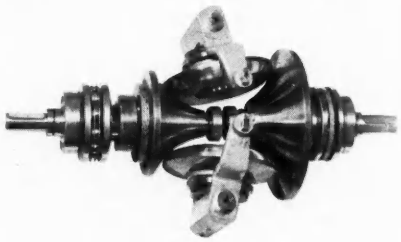


Fig. 51. Working parts of the Arter variable-speed gear.
(Courtesy of J. Arter & Co.

These units can also be supplied with a built-in differential gear, giving an infinitely variable speed range from 1·5 or 1·1 times the input speed down to zero.

The RF Continuously Variable Gear:—As can be seen from Figs. 52 and 53, the Rennerfelt RF variable-speed gear, like the Arter gear, operates on the tilting-roller principle, a number of rollers (5) serving to transmit power from a toroidal driving disc (4) to a driven disc (6). Torque is transmitted to disc (4) through one or more driving pins (3) from a gear-wheel (2) meshing with a motor pinion (1). On the shaft (7) is mounted a sleeve (8), which, by means of one or more driving members (9), is non-rotatably connected with disc (6). At some distance from sleeve (8) is mounted another sleeve (10) keyed to the shaft (7). Between the sleeves and entering into V-shaped recesses are balls (11). When disc (6) rotates and load is applied, the two sleeves (8) and (10) will be forced apart.

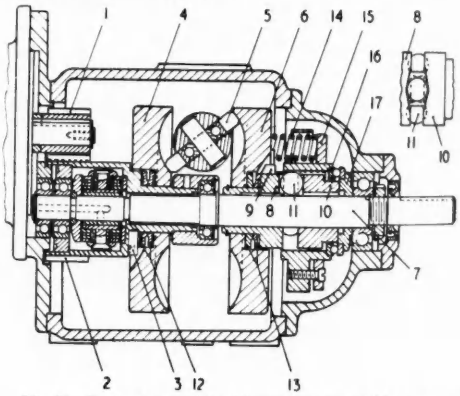


Fig. 51. Sectional view of the RF continuously variable gear.
(Courtesy of Brottby Mekaniska Verkstads AB.)

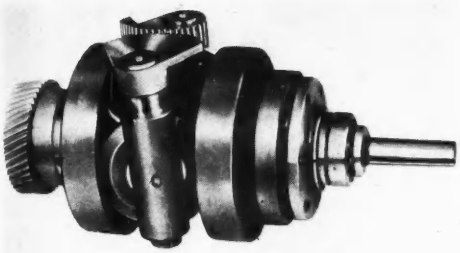


Fig. 53. **Working parts of the RF continuously variable gear.** (*Courtesy of Brottby Mekaniska Verkstads AB.*)

In order to compensate for minor irregularities in the co-operating surfaces of discs (4) and (6) and the intermediate rollers (5), these discs are mounted on their shafts with a certain degree of play, allowing for a minor degree of adjustment, made possible by the spring washers (12) and (13).

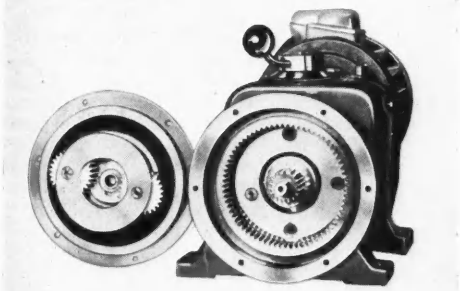


Fig. 54. **RF variable-speed drive with planetary gear.** (*Courtesy of Brottby Mekaniska Verkstads AB.*)

This arrangement provides for an axial pressure between the discs, the magnitude of which varies in accordance with the speed of the driven shaft (7), the pressure being greatest at the lowest speed of this shaft. If the power transmitted is constant and the speed of the driven shaft (7) increases, the axial pressure decreases with the square of the speed, whereas theoretically the pressure should decrease linearly. In order to provide for better agreement between the desired pressure and that obtained, a number of springs (14) are distributed over disc (6) and are held between the disc and a ring (15), which, through a thrust bearing (16), bears against the intermediate member (17).

The device comprising the sleeves (8) and (10) and the balls (11) is difficult to manufacture with such precision that the same load is imposed on all the balls. To overcome this drawback, sleeve (8) is designed to move to some extent in a plane at right angles to shaft (7), at the same time transmitting the torque from disc (6) through the balls (11) and the sleeve (10) to the shaft (7). Thus, the arrangement automatically adjusts itself to such a position that all the balls (11) share in the transmission of the torque.

These Swedish-built gears can be obtained in capacities from 1 to 25 hp, with a speed ratio of 6:1, maximum output speeds being equal to the motor speed. Smaller sizes, ranging from ¼ to 2 hp, are available in combination with planetary gears for low speeds, a typical example being shown in Fig. 54. In this case, the major part of the motor power is transmitted directly through the sun gear and a minor part through the stepless gear, output speeds being from zero to 100 rpm or less, according to the gear ratios installed. The efficiency of the RF gear ranges from 80 to 95%.

The Hayes Gear:—Whilst the policy of this survey is to describe only those variable-speed drives which are readily available commercially, it is felt that an exception may profitably be made in the case of the Hayes gear. This gear will be remembered by automobile engineers as having been installed in Austin cars between 1934 and 1938. It is odd that the Hayes gear, which undoubtedly represents one of the most advanced forms of friction drive, should have fallen into virtual oblivion. At present, however, it is being revived by Tiltman Langley Ltd. as a possible alternator drive system for aircraft. Although, in this particular application, constant output is required from a variable input speed, it is quite possible that the Hayes gear may be adapted to industrial drive applications. In this connection, therefore, the following description is not of academic interest only.

The principle of the Hayes gear (Fig. 55) involves the use of toroidal-shaped discs (1), (2), and (3), with rollers (4) loaded between the discs and capable of being rotated into various angular positions to change speed ratio. As shown in Fig. 55, the transmission actually comprises two stages driven in parallel, and thus, with three rollers in each stage, there are six rollers transmitting. In order to load the gear so that it can transmit torque, a cammed ring (5) on the input side transmits the drive via three balls (6) to the discs and, by virtue of its cam form, imposes an axial thrust, depending on the torque transmitted. The rollers are housed in carriages (7), which are in turn supported by rockers (8) on a spider (9), and which can swivel in these rockers because of the ball ends attached to the carriages and the seatings provided in the rockers.

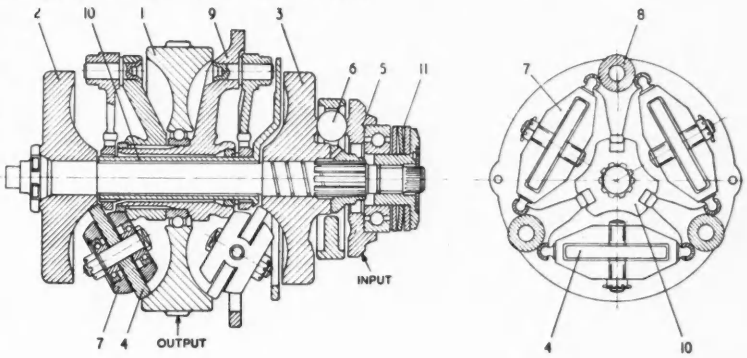


Fig. 55. **Cross-section through the Hayes gear.** (*Courtesy of Tiltman Langley Ltd.*)

The rockers themselves are connected to a central control tube (10), which is free to float both axially and radially. Rotation of this control tube causes the rockers to move the carriages in such a way that the rolling axes of the rollers are no longer coincident with the axes of the discs. Consequently, a side load bears on the rollers, which then rotate on the rocker seatings until the roller axes of rotation again become coincident with the axes of the discs. This is effected by setting the roller carriages (7) in such a way that a line connecting the centres of the carriage ball ends is not normal to the transmission centre-line but slightly at an angle to it. As a result, quite a small angular movement of the central control tube (10) can accomplish an effective ratio change. It also provides for complete equalization between the rollers, since the reaction forces on the rocker connections to the control tube can be utilized to permit automatic adjustment. The control tube is capable of radial movement, and thus, if the forces at these points are unequal, the tube will adjust itself so that they equalize. The control-tube members engaging the rockers are angled in opposite directions, so that the axial thrusts produced ensure a similar equalization between the stages in the transmission. Initial axial loading is provided by Belleville springs (11).

While, as stated above, the Hayes gear is being developed for aircraft applications involving a constant output of 30 kW at 8000 or 12,000 rpm, with inputs varying from 3000 to 10,000 rpm, industrial adaptations of this drive will probably be suitable for applications up to 15 hp at correspondingly lower speeds. It is expected that, in such cases, the available speed ratio will be 4:1.

The Metron Miniature Variable-Speed Changer:—The Metron miniature variable-speed changer also employs the toroidal-disc and roller principle but, in contrast with the designs previously described, is intended for extremely low-power applications, e.g., in cardiographs, computers, optical drives, oscillographs, timers, servomechanisms, and process controllers. As can be seen from Fig. 56, this unit is of extremely neat and compact design, the maximum horsepower output being 0·025 hp and the rated output torque from 5 to 40 oz-in., depending on the output-speed setting. Rotation can be effected in either direction, the maximum speed of either shaft being 10,000 rpm. Speed ratio is from 1:5 up to 5:1 down, giving an overall speed ratio of 25:1. Any nominal output speed is possible by incorporating, if required, one of a range of built-in fixed-ratio speed changers. In addition, a wide choice of speed control is available, including operation by lever, by push-rod, and by spur, mitre, or worm gears.

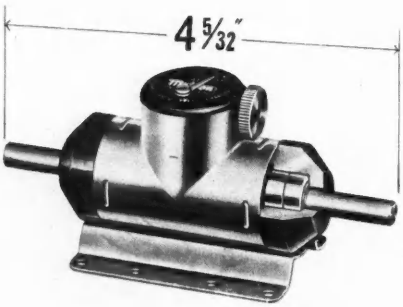


Fig. 56. The Metron miniature variable-speed changer. (*Courtesy of Metron Instrument Company*)

The Kopp Variator:—Invented by the Swiss engineer, Jean E. Kopp, this variable-speed device is now manufactured by eight companies all over the world, and is generally recognized as one of the most successful designs of the friction type. The principle of operation is illustrated in Fig. 57. When the cones act on equal diameters of the balls (Fig. 57a), input and output speeds

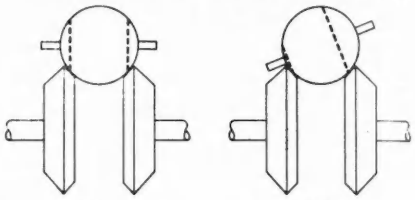


Fig. 57. Principle of operation of the Kopp variable-speed gear.

are also equal ; when, however, the axes of the balls are tilted (Fig. 57b), one cone will operate on a reduced ball diameter, and the other on an increased ball diameter. The balls can be tilted in either direction, giving an overall infinitely variable ratio of 9:1. An exploded view of the Kopp drive is shown in Fig. 59. The centrifugal forces of the balls (1), of which there are from three to eight, according to the manufacturer and the horsepower to be transmitted, are taken up by a retaining ring (2). These balls are drilled through their centres to receive spindles (3), which project on both sides and are guided in radial slots in the covers (4) and (5). At the same time, they pass through spiral slots in a speed-adjusting plate (6), which can be turned inside the housing by a projecting lever arrangement (7). In some, particularly the large, units, the speed-adjusting plate may be operated by a handwheel-actuated worm gear, as shown in the inset. Contact forces are controlled by a special torque-responsive mechanism, so that the pressure required to transmit a given torque is automatically produced.

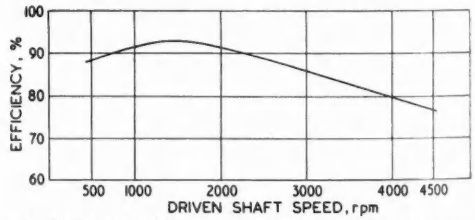


Fig. 58. Typical efficiency curve of the Kopp variable-speed gear. (*Courtesy of Allspeeds Limited*)

Kopp units are normally available in capacities from as low as 1/33 hp (weighing as little as 14 oz) to 14 hp, and are rated for full power at all speed settings. There are, of course, many applications, e.g., in machine tools, in which this feature is particularly desirable. The speed ratio, as mentioned above, is 9:1, and the speed range from one-third to three times the input speed (usually 1500 rpm). Slip is relatively small, but may reach 4% at the lowest output speed. A typical efficiency curve is shown in Fig. 58.

These units operate with a minimum of noise and

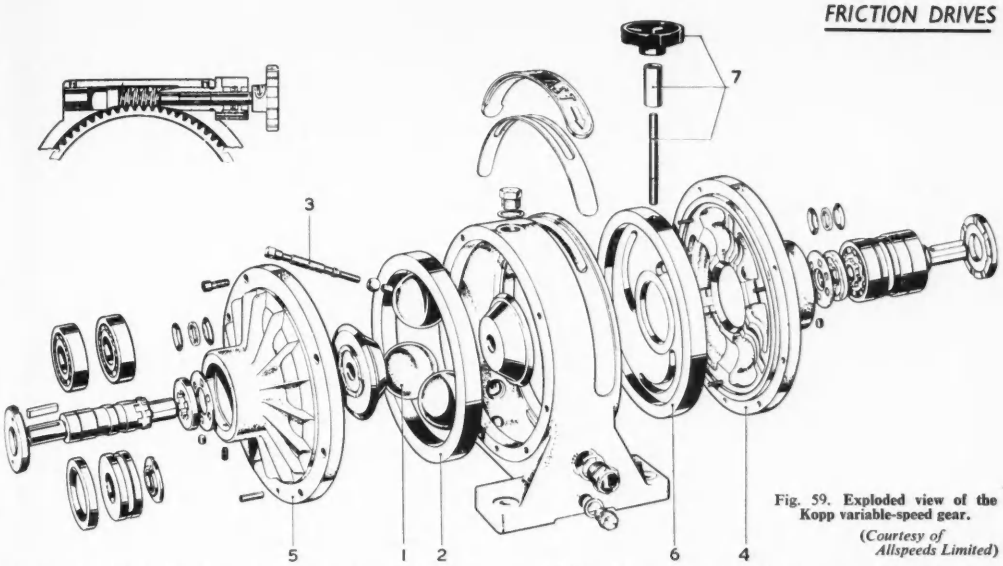


Fig. 59. Exploded view of the Kopp variable-speed gear. (*Courtesy of Allspeeds Limited*)

vibration in either direction, and can reverse instantly, even under full load. As with most other friction drives, they can be supplied with or without motor and with a built-in gearbox for shifting speed range, as required.

The Contraves Speed Variator:—Like the Kopp, the Contraves variable-speed drive uses steel balls as the power-transmitting elements. Its mode of operation can be seen from Fig. 60. Four balls (1) are pressed against ball races (4) and (5) by means of an automatic tensioning device (2), these races being mechanically coupled to the driving and driven shaft respectively. Speed variation is effected by rotating the four control rolls (6), which adjust the position of the axes of rotation of the balls (1). This is made possible because the contact pressure between the balls (1) and rolls (6) is higher than the respective pressures of races (4) and (5) against balls (1). The position of rolls (6) is changed by means of worm gearing from a control spindle (3), which projects through the hollow input shaft.

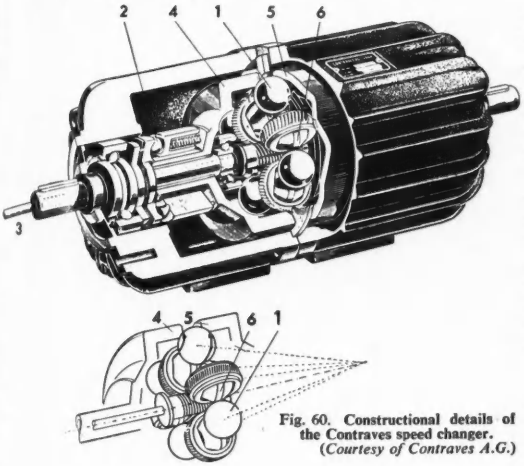


Fig. 60. Constructional details of the Contraves speed changer. (*Courtesy of Contraves A.G.*)

Whilst Contraves speed changers are built for low horsepowers only, about ¾ hp being the maximum, their speed ratio is exceptionally high, i.e., 1:2 when increasing with respect to the input speed (2000 rpm maximum), and 10:1 when reducing, giving an overall ratio of 20:1. The drives are rated at constant horsepower in the high-speed range, and constant torque in the low-speed range ; the horsepower characteristic is thus of the type shown by curve (*c*) in Fig. 1.

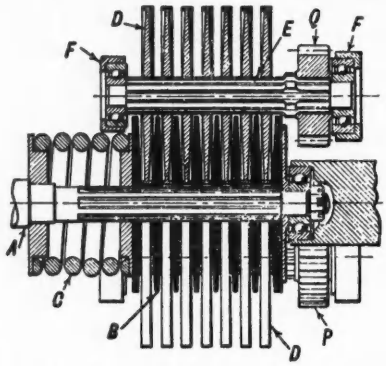
Incidentally, it is interesting to note that, as far as can be judged from various Russian publications, a ball-type speed changer, which, to all intents and purposes, seems to be an exact copy of the Swiss-built Contraves drives, is marketed in the U.S.S.R., where it appears to enjoy considerable popularity. The Russian unit is available in five sizes, ranging from about ¼ to 3 hp, with a speed radio of 1:2 up and 5:1 down, giving an overall speed ratio of 10:1.

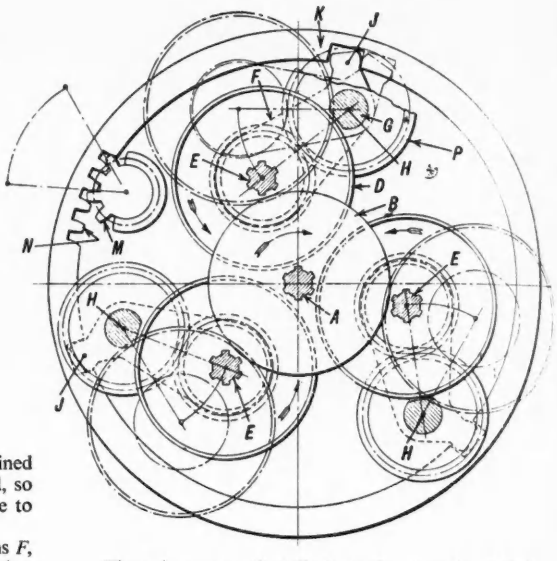
The Beier Infinitely Variable Gear:—The important principle of the Beier gear, which is basically different from that of other friction drives, is that power is transmitted through the drag of thin films of oil interposed between driving and driven members. Thus, actual metal-to-metal contact, and consequently wear, of the power-transmitting surfaces are prevented.

The principle of operation can be seen from Fig. 61. Tapered discs *B* are carried on a splined shaft *A*, on which they are free to slide axially. At the same time, they are pressed against flanged discs *D* by a spring *C*. Sets of flanged discs *D* are mounted on three splined shafts *E*, on which they are likewise able to move axially. Output speed is varied by changing the depth to which the flanged discs *D* are inserted between the tapered discs *B*, thereby altering the effective diameter of the tapered discs, whilst that of the flanged discs remains constant.

By using several sets of flanged discs equally spaced around the tapered discs, these are maintained accurately at right angles to their shaft and are prevented from tilting. The flanged discs are also held at right angles to their respective shafts, partly by centrifugal force, and

Fig. 61. **Design features of the Beier infinitely variable gear.** (*Courtesy of Beier Infinitely Variable Gear Co. Ltd.*)





partly by the pressure of the tapered discs. The splined holes through all the discs are slightly bell-mouthed, so that, even if they should tilt a little, they are still able to slide freely on their shafts.

The ends of shafts *E* are carried in swinging arms *F*, which are mounted on pins *G*, pivoting about *H*, so that the position of the flanged discs is altered with respect to the tapered discs. Three projections *J* of the arms engage with recesses in the common adjusting ring *K*, which can be rotated in the gear casing by a pinion *M*, engaging with gear segment *N*, so as to change the speed ratio. The swinging arms can also be interconnected by push-pull links, as shown in Fig. 62. Power is transmitted through the gears *Q* to idlers *P* mounted on the pivot pins *Q*, and these, in turn, drive a common central gear-wheel (not shown in Fig. 61). Input and output shafts are thus coaxial. The gear can operate in either direction and can be driven from either side.

There is no practical limit to the capacity of Beier gears. Standard designs range from ½ to 300 hp at 1500 rpm input; with special designs, as much as 1000 hp can be transmitted. These high capacities can be achieved by virtue of the fact that with, say, three sets of nine flanged discs, each with two sides, there are 54 points at which power is transmitted. This number can be augmented, if necessary, by increasing the number of discs on each splined shaft, by increasing the number of splined shafts (up to nine in certain cases), and by increasing the diameter of the discs, so as to obtain a greater mechanical advantage. However, up to about 40 hp, three sets of flanged discs are sufficient. In the 300-hp

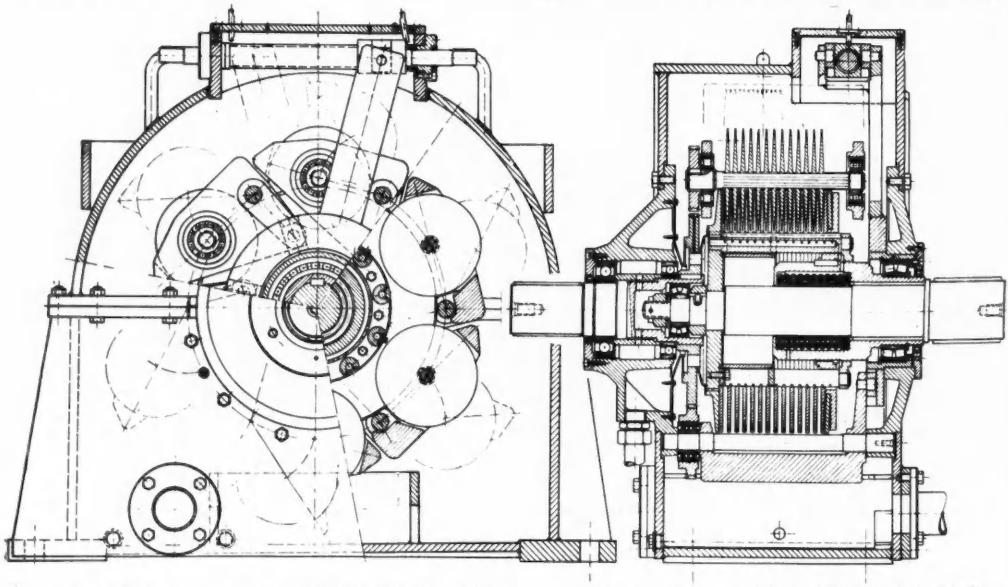


Fig. 62. **Sectional arrangement of 300-hp Beier infinitely variable gear.** (*Courtesy of Armstrong Whitworth (Metal Industries) Ltd.*)

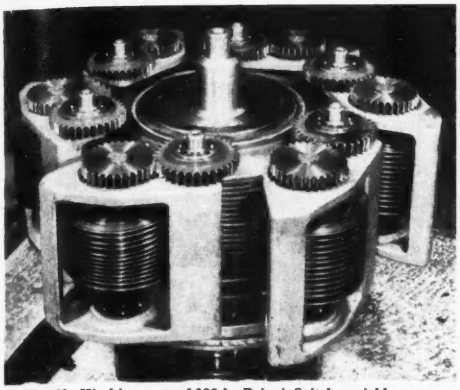


Fig. 63. **Working parts of 300-hp Beier infinitely variable gear.** (*Courtesy of Armstrong Whitworth (Metal Industries) Ltd.*)

transmission shown in Figs. 62 and 63, six sets are used, the flanged discs being mounted on the central shaft, and the tapered discs on the swinging units. Efficiency curves for a 300-hp and for a 15-hp transmission are given in Fig. 64. Speed ratios of standard designs vary from 3·5:1 to 5:1; input speeds range from 750 to 3000 rpm.

Under normal operating conditions and at high output speeds, the gear is capable of transmitting its rated power with negligible slip ; in the low-speed range, however, slip may be appreciable. Should this be objectionable, a special slip-compensating device may be incorporated, so that changes in load will have no effect on output speeds.

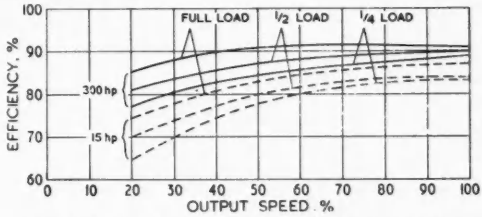


Fig. 64. **Efficiency curves for a 300-hp and a 15-hp Beier gear.** (*Courtesy of Armstrong Whitworth (Metal Industries) Ltd.*)

Three types of gears are available, i.e., with constant-horsepower rating, with constant-torque rating, and with an intermediate rating in which horsepower capacity at low output speed is somewhat higher than with the constant-torque type. Constant-horsepower types may be so arranged that output speed is adjusted automatically, according to the output torque or any desired load-speed characteristic. Such wide performance variations are made possible by the great flexibility in the design of these gears, which are now being built by several manufacturers and which, incidentally, are also being tested in Germany as automobile transmissions.

IMPULSE DRIVES

The fundamental difference between impulse drives and belt and friction drives (with the possible exception of the P.I.V. gear described on page 53) is that power transmission is entirely positive. The heart of impulse drives consists of some kind of variable-stroke mechanism acting on free-wheel (one-way) clutches. Whilst these drives are not nearly as well known and as frequently employed as belt or friction drives, they have their own fields of useful application, particularly in low-horsepower requirements with high speed ratios.

The Morse Variable-Speed Control:—The operating principle of the Morse gear is best explained by reference to Figs. 65 and 66. The input shaft is integral with a dual-cheek eccentric (1), which causes the three sets of idler links (2) to oscillate about their pivots (3) in the lugs (4) of the adjusting ring, which is rotatable by a handwheel and worm gearing (8). The angle of swing of the links is constant and is determined by the fixed offset of the eccentric. Three sets of one-way clutches and their appropriate linkage mechanisms are fitted. By adjusting the centre distance between the axis of each clutch (5), and the pivots (3) of the idler links (2), the amount of movement transmitted to the arms (6), which are integral with the clutch shaft and hub, by the drag links (7) can be varied. Movement of the pivot away from the clutch axis increases the movement transmitted to the clutch arm. When the pivot nears the clutch axis, the linkage " jack-knifes ", and the drag links impart only a small oscillation to the clutch.

The internal mechanism of the clutch (spring-actuated cams) transmits only the forward motion of the clutch through the clutch gear (9) to the output gear (10), which is keyed to the output shaft. The return motion of the clutch is made as the clutch overruns in this direction. Because of their spacing around the unit, the three clutches operate successively, one performing the forward stroke, whilst the other two free-wheel.

Since the cams in the clutches are highly loaded when the clutch is engaged, and then unstressed during the return stroke, a fatigue problem exists. The life of the clutch and its ability to function properly are dependent on the loads encountered in operation and on the number

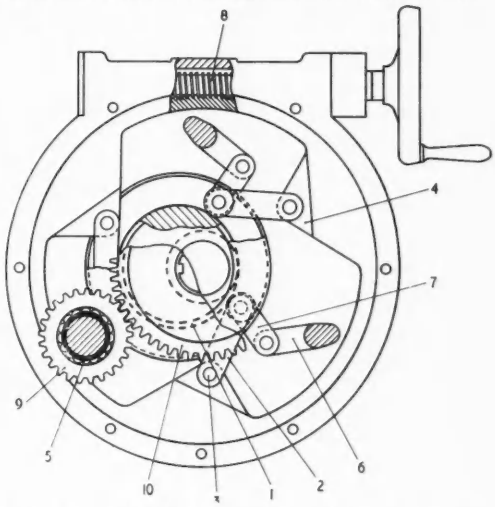


Fig. 65. **Operating principle of the Morse variable-speed control.** (*Courtesy of Morse Chain Co.*)

of stress applications. A small reduction in operating load will effect a large increase in the number of engagements which the clutch will withstand before fatigue failures are encountered. It is therefore advisable to inspect the clutches at regular intervals.

The maximum input speed of the Morse gear is 250 rpm, with output speeds ranging from 55 down to 2 rpm. Maximum torque at the output shaft is 200 lb-ft and this is constant over the whole speed range. Efficiency is at least 95% at any speed setting.

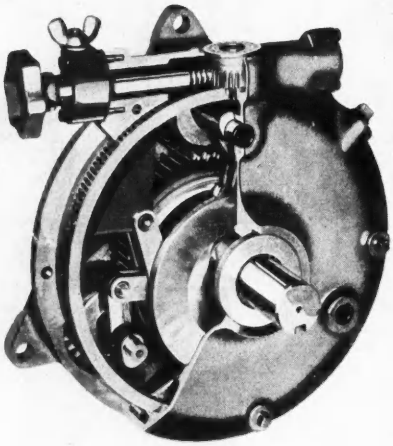


Fig. 66. Cut-away view of the Morse variable-speed control. (*Courtesy of Morse Chain Co.*)

Owing to the mechanical action within the unit, there is a slight pulsating movement transmitted to the output shaft. In many applications, e.g., feeders and mixers, this pulsating motion may be a distinct advantage.

The Zero-Max Transmission:— In the Zero-Max transmission, power transmission is effected by several sets of linkages, mounted side by side and operated in phased sequence. As can be seen from Fig. 67, the input shaft *A* carries eccentrics *B*, which transmit a lateral oscillatory motion to the connecting links *D*, via the connecting rods *C*. This oscillatory motion is converted into a rotary motion of the output shaft *E* through one-way clutches *F*. The output speed is adjusted by changing the position of the pivot of control links *G*. Speed regulation down to zero is possible, in which case the pivot of control links *G* is in such a position that the connecting rods *C* merely move up and down, and no lateral motion is imparted to the connecting links *D*. When the control links *G* are moved away from the zero position, the stroke of the connecting links *D* becomes longer, and output speed is increased. This transmission can be operated in the zero position without any generation of heat, and can thus carry out the function of a clutch.

These units transmit constant torque over the whole speed range, ratings being from 10 to 100 lb-in. The

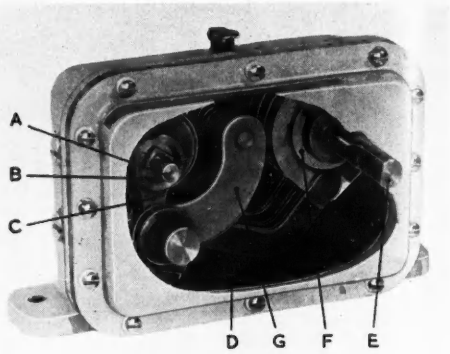


Fig. 67. Cut-away view of the Zero-Max gear. (*Courtesy of Revco Incorporated.*)

output-speed range is from 0 to 25% input speed in standard models, but reversing models are also available up to a maximum of 20 lb-in., with an output which is infinitely variable to 25% input speed both sides of zero. Output speed is not strictly uniform, particularly near zero output—a feature common to all impulse types of transmission. However, as in the case of the Morse gear, this impulse feature will usually be of no importance, as the inertia of the driven machine will help to smooth out irregularities.

The illustration in Fig. 67 shows a model in which speed changes are effected by means of a lever control. Where greater accuracy is required, a handwheel-and-screw control can be supplied as optional equipment.

The GUSA Infinitely Variable Speed Drive:— Unlike the impulse drives just described, the GUSA gear is built with capacities ranging from 1 to 10 hp. As shown in Fig. 68, the crankshaft (1) operates the connecting rod (2), which slides in a sleeve (3). This sleeve is itself designed to slide in a rocker bearing (4), carried by a crosshead (5). A lever arm (6) is connected to sleeve (3) through a pin (7) and operates a one-way clutch (8) fitted to the output shaft. Pin (7) thus describes an arc about the axis of the one-way clutch (8), the angle enclosed by this arc being determined by the position of crosshead (5). Operation of a screw (9) by means of a handwheel (10) will move the crosshead (5) towards or away from the one-way clutch (8), and will thus effect speed changes. Zero output speed is obtained when

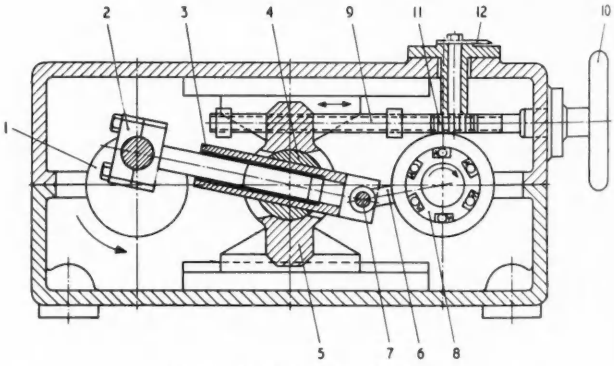


Fig. 68. Sectional drawing of GUSA gear. (*Courtesy of Heinrich Gensheimer & Söhne*)

pin (7) coincides with the centre of rocker bearing (4). A speed dial (12) is operated via a pinion (11) from screw (9).

In this arrangement, 200 to 215 deg. of crank rotation are utilized for the working stroke, while 160 to 145 deg. correspond to the return stroke, during which the one-way clutch free-wheels. It is clear, therefore, that, if two such drive arrangements are used with a relative displacement of 180 deg., the active parts of the cycles of each mechanism will overlap. In the actual construction of these drives, however, three mechanisms are operated from the same input shaft, with their cranks at an angle of 120 deg. relative to each other, thus ensuring a smooth output speed. The three one-way clutches on the output shaft are therefore operative with a considerable amount of overlap.

Fig. 69. View of typical GUSA gear. (*Courtesy of Heinrich Gensheimer & Söhne.*)

These drives are made in five models, i.e., 1, 2, 3, 6, and 10 hp. For the 1- and 2-hp models, the speed range is from 150 rpm down to zero, with a constant-horsepower characteristic from 150 to 110 rpm, and a constant-torque characteristic from 110 rpm to zero. The 3-, 6-, and 10-hp drives give a speed range from 300 rpm to zero, with constant horsepower down to 150 rpm, and constant torque thereafter. The input speed in all models is 1500 rpm and the efficiency better than 80%. A photograph of a typical GUSA gear is shown in Fig. 69.

Conclusion

An attempt has been made in this survey to indicate the scope and application of mechanical infinitely variable speed drives, and to outline the major points influencing their selection in different industrial fields. While it is, of course, impossible in an article of this nature to refer to every make of speed changer in existence, descriptions have been given, to the best of the author's knowledge, of all the major types in use to-day. In this connection, it may be argued that some of these descriptions have perhaps included a considerable amount of technical detail which at first sight may not appear strictly necessary to prospective purchasers, who, after all, are probably more interested in the actual performance of a machine than in its method of operation. It should be understood, however, that most variable-speed applications have their own particular problems, and this article is not intended to supplant prior consultation between prospective users and manufacturers of such equipment. On the contrary, the fullest co-operation between purchaser and manufacturer is absolutely essential, if optimum results are to be achieved. Therefore, some knowledge of the working principles of all types of drives is desirable and, by including these descriptions, purchasers will have the opportunity of acquiring a wider knowledge of variable-speed drives in general and will be better equipped to judge the validity of the claims, competitive or otherwise, put forward by individual manufacturers.

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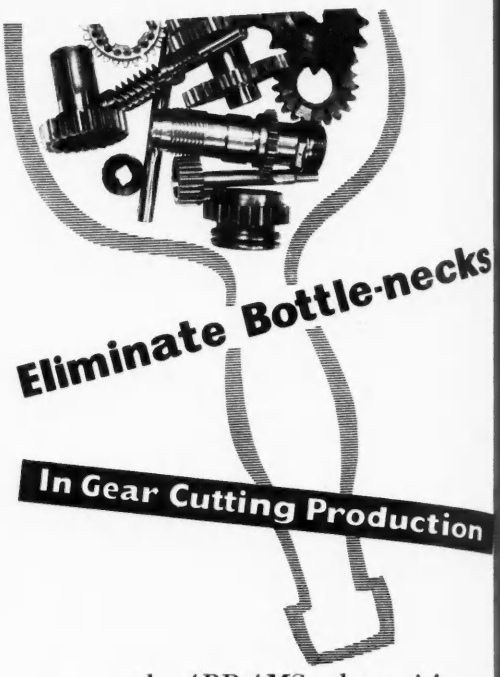

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THE ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. XVIII No. 2 FEBRUARY, 1957

Applications of Oil-Mist Lubrication in the Steel Industry

By T. M. MURPHY. (From *Iron and Steel Engineer, U.S.A.*, Vol. 33, No. 12, December 1956, pp. 77–81, 4 illustrations.)

OIL-MIST lubrication may be defined as a method of converting liquid oil into an airborne mist which may be conveyed in an air stream through a system of piping to a number of bearings for the purpose of lubrication. The past five years have seen many successful applications of the oil-mist method of lubrication. However, numerous enquiries and reports from the field on its ever-widening application indicate that a review of the oil-mist principle of lubrication is in order.

The first applications of an oil-mist principle of lubrication were utilized when an air-line oiler was applied to the air lines supplying pneumatic tools. However the oil mist generated by an air-line oiler varied uncontrollably in oil-mist particle size. In fact, 94 to 96% of the mist generated was of so large a particle size that it readily collected on the walls of the pipe or tubing. This fact prevented equitable or controlled distribution of the oil to more than one point of application. However, present-day oil-mist lubricator design has eliminated this problem, and now about 96% of the mist generated has a particle size of two to five microns in diameter, enabling it to be distributed and controlled.

The primary components of the oil-mist lubricator are an oil reservoir, an air-supply inlet, an atomizing nozzle, a baffle plate, and an oil-mist outlet. The amount of oil passing through the oil-mist lubricator at 20 psi is approximately 1·2 cfm. The amount of mineral oil with an average viscosity of 500 saybolt seconds at 100°F being dispersed at 75°F into the air is 0·01 cu in., or approximately two drops of oil per cubic foot of air. The mixture, therefore, is not sufficiently concentrated to become inflammable. The mixture can be conveyed through ½-in., or larger, pipes for recommended distances of 30 ft in any direction from the lubricator, without being affected to any marked extent. In tests it has been successfully passed through as much as 100 ft of tubing. However, if the oil mist is passed through small bores, even for short distances, the amount of precipitation or condensation will be considerable.

Almost any liquid with a viscosity not exceeding 500 to 600 saybolt seconds and not more volatile than kerosene or very light lubricating oil can be misted. Attempts have been made to mist water, but these have been unsuccessful, though some satisfactory misting has been accomplished with soluble oil in water.

The fact that oil can be converted to mist and conveyed reasonable distances through pipes of suitable size, without any marked precipitation effect, and then can be reconverted to liquid oil at will by passing for short distances through small-bore holes, gives considerable control over its application for the solution of lubrication problems. Thus, the oil mist can be applied in three ways, i.e., in the form of a straight mist as generated, as a semi-condensed mist by passing it through spray fittings of a reduced bore, or by passing it through fittings of very small bore, condensing approximately 90% of the mist back to droplets of liquid oil.

The oil-mist method has proved to be a good means of lubrication between balls or rollers and their retainers, cages, and races. It assists in dissipating frictional heat, while the internal pressure of the oil-mist air in the bearing housing helps to prevent the entry of dirt and other abrasive materials. Furthermore, fresh, cool lubricating oil can be applied continuously in small quantities.

Oil-mist lubrication has been successfully applied to many plain bearings, gear trains, chains, etc. However, frictional-type bearing surfaces must have full film lubrication, and it is important that such a film can be developed and maintained for each application.

To date, oil-mist lubrication has been successfully used in general industry. Insofar as steel-mill equipment is concerned, the application of the oil-mist method has replaced grease lubrication on both single stands and four-high stands in a number of steel mills. Oil mist is applied through the normal grease-inlet ports of the anti-friction bearings after removing the grease lubricant. Examination of these bearings after a full year of operation has proved that oil-mist lubrication is very satisfactory—a fact which is of great importance in mills using roll-oil for steel rolling, because of factors other than lubrication itself. Actually, it has been found that the cost of lubrication has been reduced to about 1% of its previous amount.

With the grease-lubrication method, rolling oils frequently become contaminated with excess grease leaking from the bearings. Since the application of oil mist, this contamination of rolling oils has been practically eliminated. For example, it has been found that, when the work-roll and back-up roll bearings were lubricated with grease, the rolling-oil solution had to be replaced on an average eighteen or nineteen times over a period of six months. With oil-mist lubrication, however, rolling-oil changes were reduced to approximately nine or ten over the same period, representing a very considerable saving in the cost of roll-oil solution. In addition, lubrication time was immediately reduced, and the operating temperature of the bearings was lowered.

The application of oil-mist lubrication has been very successful on steel-mill tinning lines, where it has been found to provide adequate lubrication for the bearings at a minimum cost, from the points of view of both lubricant consumption and application. However, outstanding economy has resulted from the prevention of contamination of the tinning solution and of staining of the steel. In addition, it should be noted that oil-mist lubrication has reduced the frictional drag on the rollers, particularly the idler rolls. On one particular set-up, the idler rolls were normally driven by the strip; however, the grease-lubrication auxiliary drives had to be applied

(*Concluded on page 70*)

RUSSIA

Boring Bars with Improved Resistance to Vibration

By L. K. KUCHMA. (From *Vestnik Mashinostroyeniya*, No. 9, September 1956, pp. 33-39, 13 illustrations.)

SELF-EXCITED vibrations arising during the cutting of metals have either one or two frequencies close to the natural frequency of one of the elements in the elastic system, which consists of the machine, the work, and the tool. When using a cutting tool carried by a cantilever boring bar and also when an additional steady is employed, the frequency most commonly observed in horizontal boring mills is either that of the system comprising the boring bar and its supports (low-frequency vibrations) or of the system comprising the cutting tool and tool-holder (high-frequency vibrations). Both types of vibration can sometimes be observed simultaneously.

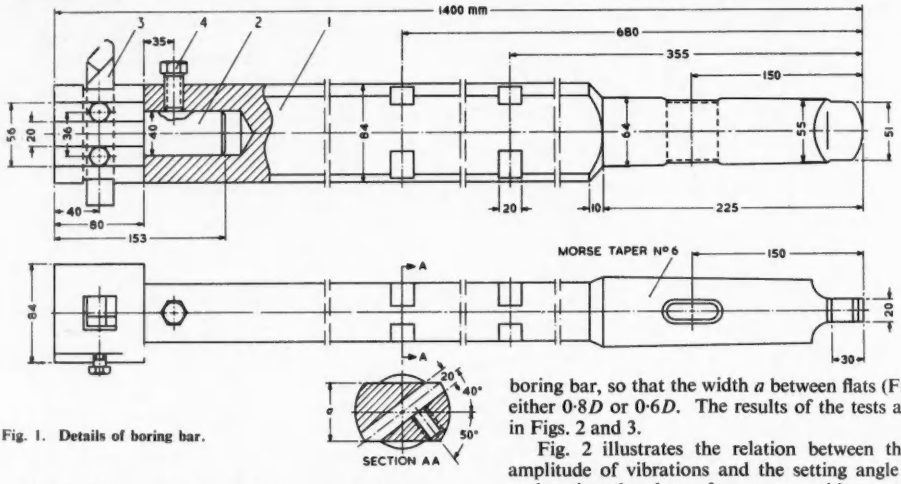


Fig. 1. Details of boring bar.

Earlier theories of tool chatter ascribed the predominant role in the formation of self-excited vibrations to the "negative characteristics", these characteristics signifying a decrease in cutting force with an increase in cutting speed. Subsequent observations have established that a system under consideration cannot be treated adequately as one with a single degree of freedom, the motion of the system being substantially determined by cross-links between the co-ordinates. In a system with two degrees of freedom and respective stiffnesses k_1 and k_2, analysis of the expression for the stability limit shows that this limit, at which self-excited vibrations arise, depends on the setting angle θ in relation to the plane of symmetry of the boring bar, and will increase with increased values of $k_2 - k_1$. Consequently, it should be possible to control the initiation of self-excited vibrations not only by altering cutting conditions and by the stiffness and mass of the elements in the vibrating system, but also by changing the direction of action of the cutting force in relation to the vibrating system with several degrees of freedom. Moreover, control can also be achieved by varying the stiffness ratios.

To apply these principles in practice, one obvious method, in which different stiffness values in the different degrees of freedom can easily be varied, is the use of a boring bar with a non-circular cross-section. The design of such a boring bar, as used in these investigations, is shown in Fig. 1. At the end of the bar (1) an orientatable cutting head (2) is mounted, with a boring-tool bit (3) which can be clamped in any desired position by means of the screw (4). Angular position is measured in relation to the plane of symmetry parallel to the flats of the bar cross-section. A bearing support was arranged between the headstock and work-table of a horizontal boring mill and, by adjusting the distance between the headstock and bearing support, the cantilever length of the boring bar could be varied at will. Variations in the stiffness ratio k_1/k_2 were effected by changing the size of the flats.

Tests were carried out with flats machined from the round along the entire length of the bar to a depth of either $0{\cdot}1D$ or $0{\cdot}2D$, where D is the diameter of the boring bar, so that the width a between flats (Fig. 1) was either $0{\cdot}8D$ or $0{\cdot}6D$. The results of the tests are shown in Figs. 2 and 3.

Fig. 2 illustrates the relation between the double amplitude of vibrations and the setting angle θ of the tool against the plane of symmetry, with a cutting speed of 40 m/min, a feed of 0·3 mm/rev., and a depth of cut of 3·0 mm. In these tests, the length of the cantilever portion of the mandrel was maintained constant at 550 mm, the flats being machined to a depth of $0{\cdot}1D$. A

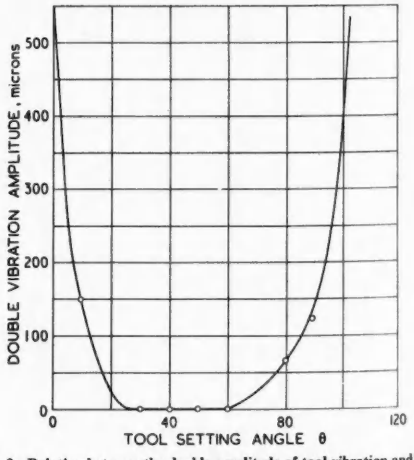


Fig. 2. Relation between the double amplitude of tool vibration and tool setting angle θ.

right-hand turning tool was used, with a front rake of 10 deg., main and auxiliary cutting-edge clearance angles of 8 deg., a main cutting-edge angle in plan view of 60 deg., an auxiliary cutting-edge angle of 70 deg., a main cutting-edge slope of 0 deg., and a cutting-tip radius of 0·5 mm. This cutting-edge geometry was maintained in all other tests. It can be seen from Fig. 2 that, with the cutting tool set to an angle θ between about 20 and 60 deg., cutting proceeded smoothly without vibration. However, with lower or higher values of θ, intensive vibration was produced, with double amplitudes reaching more than 500 microns.

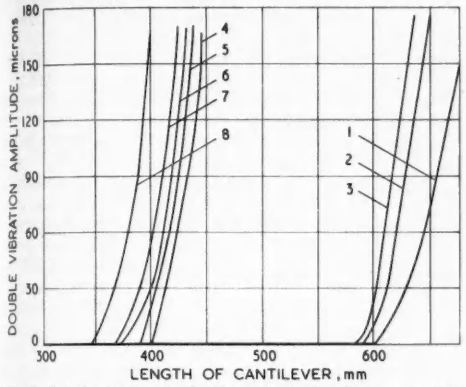


Fig. 3. Relation between the double amplitude of tool vibration and the length of the cantilever portion of the boring bar at different tool setting angles θ.
(1) θ = 40° ; (2) θ = 30° ; (3) θ = 50° ; (4) round boring bar ; (5) θ = 130° ; (6) θ = 170° ; (7) θ = 110° ; (8) θ = 150°.

Fig. 3 shows the relation between the double amplitude of vibration and the length of the cantilever portion of the boring bar at different setting angles θ of the tool, using right-hand tool bits and similar cutting conditions as before, with the exception that the depth of cut was reduced to 2·0 mm. These tests were carried out with a boring bar having flats cut to a depth of 0·1*D*, and also with a round bar. Good results were obtained with tool setting angles θ between 30 and 50 deg., the optimum value being 40 deg. Other setting angles showed very low resistance to vibration, the lowest resistance being developed at 150 deg. With a round boring bar, equally low resistance to vibration was observed, of the same order as that of the bar with flats at its worst setting angle.

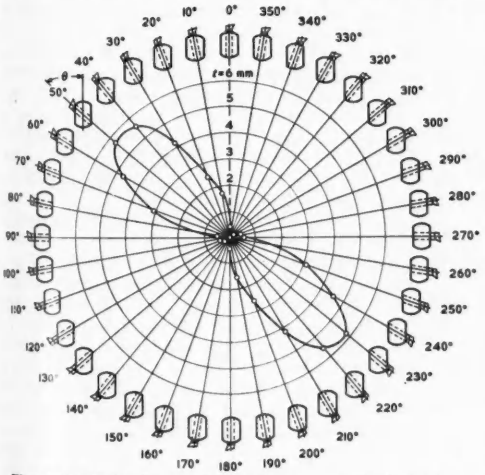


Fig. 4. Graph showing the relation between permissible depth of cut and tool setting angle θ.

Fig. 4 shows the permissible depth of cut *t*, below which vibrations do not arise, for different setting angles θ of the tool. In these tests, the cutting speed was 40 m/min and the feed 0·1 mm/rev., using a boring bar with a cantilever length of 500 mm and a diameter of 85 mm, with flats cut to a depth of 0·1*D*. When working with right-hand cutting tools, the most favourable setting angle was found to be between about 30 to 50 deg., or between 210 and 230 deg. With the tool at its least favourable setting angle, vibrations developed with a depth of cut between 0·3 and 0·5 mm, whereas at the optimum setting angle no vibrations were observed with a depth of cut up to 6 mm.

Comparative data on the vibration resistance of the system, using a round boring bar and bars with flats, are given in Fig. 5, which shows the relation between the depth of cut *t* and the tool setting angle θ, for a round boring bar (curve (1)), and for bars with flats machined to a depth of 0·2*D* (curve (2)) and 0·1*D* (curve (3)). The cutting conditions were the same as those used in previous tests. It can be seen that the maximum permissible depth of cut *t* is about 0·3 for a round boring bar, about 2 mm for the bar with the 0·2*D* flats, and about 5·75 mm for the bar with the 0·1*D* flats. Thus, the tests establish that there is an optimum difference between the stiffness values k_1 and k_2 of the boring bar affording the highest resistance to vibration.

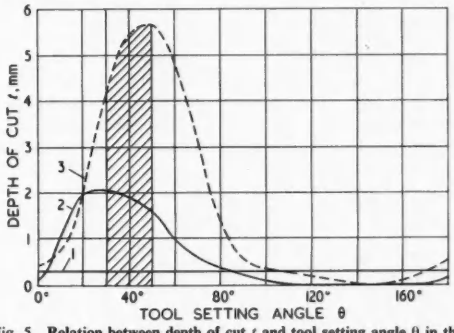


Fig. 5. Relation between depth of cut *t* and tool setting angle θ in the range between 0 and 180 deg., for a round boring bar and for bars with flats machined to depths of 0·1*D* and 0·2*D*.

It was found that variations in the diameter of the boring bar do not substantially influence the optimum setting angle θ of the tool. Thus, for example, tests with a boring bar with a diameter of 64 mm, a cantilever length of 250 mm, and flats machined to a depth of 0·1*D*, showed that the optimum setting angles for this bar were invariably equal to those with the 85-mm bar.

Tests were also carried out on a pillar-mounted workpiece, the distance between the faceplate and the pillar being 1560 mm, and between the faceplate and the cutting edge 780 mm. The length of the protruding portion of the spindle was 660 mm. With a boring-bar diameter of 84 mm and a spindle diameter of 110 mm, the average length-to-diameter ratio was 15:1. One set of tests was conducted at a constant cutting speed of 40 m/min and a constant feed of 0·3 mm/rev., with varying depth of cut, using a right-hand turning tool and either a round boring bar or a bar with flats at a tool setting angle θ of 40 deg. When working with the round boring bar, it was found

that smooth, vibrationless cutting was possible only up to a depth of cut of 3 mm, whereas a boring bar with flats enabled the depth of cut to be increased to 9 mm without any vibration whatsoever. Similar tests were also effected, using cutting tools with a special geometry, with a constant cutting speed of 40 m/min, a constant depth of cut of 0·5 mm, and a varying feed. These tests indicate that, under the conditions used, a boring bar with flats makes it possible to use a feed which is seven times greater than that obtainable with a round bar.

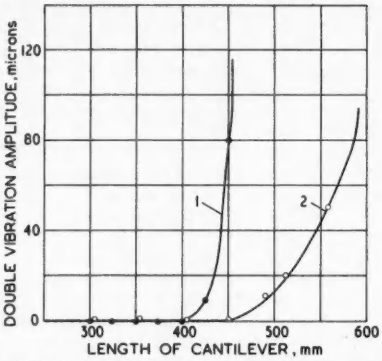


Fig. 6. **Relation between double amplitude of tool vibration and length of cantilever portion of a boring bar with flats used in a lathe, with tool setting angles θ of (1) 45 deg. and (2) 135 deg.**

As a check, tests were also carried out with boring bars used in a lathe, the bar with flats being set up in the tool support. Tool setting angles θ of 45 and 135 deg. were chosen, i.e., the most favourable and the least favourable dispositions of the cutting tool with relation to the boring-bar cross-section. Fig. 6 shows the relation between the double amplitude of vibration of the tool and the length of the cantilever portion of the boring bar, and Fig. 7 the relation between the double amplitude of vibration and the depth of cut *t*.

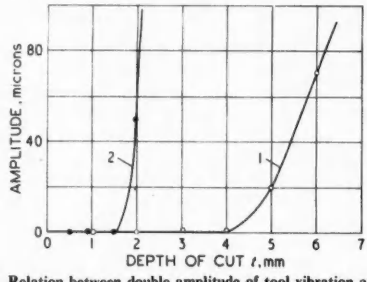


Fig. 7. **Relation between double amplitude of tool vibration and depth of cut *t* for a boring bar with flats used in a lathe, with tool setting angles θ of (1) 45 deg. and (2) 135 deg.**

The vibration-attenuation effect produced by a variable-stiffness boring bar is much smaller in a lathe than in a boring mill. This is explained primarily by the fact that the mounting of the bar in a lathe is substantially less rigid, and therefore the part played by the stiffness of the boring bar itself in the total stiffness of the system is smaller.

It should be mentioned that, when right-hand cutting tools are replaced by left-hand tools, the most favourable ranges of setting angles θ are no longer 30 to 50 deg. and 210 to 230 deg., as indicated previously, but now become 130 to 150 deg. and 310 to 330 deg.

The results of these tests show conclusively that the use of boring bars with flats, in conjunction with an optimum range of tool setting angles, considerably increases the vibration resistance of the system. The simplicity of this method of vibration attenuation is noteworthy, and the general approach to the problem, as indicated in this investigation, should enable many effective design solutions to be found for increasing the vibration resistance of machine tools.

Applications of Oil-Mist Lubrication in the Steel Industry

(*Concluded from page 67*)

to these rolls to start them in motion and this resulted in scratches on the tin-plate, causing scrap. After the oil-mist method was applied, the friction in the idler-roll bearings was reduced to the point where the rolls moved freely with the strip, and it was not even necessary to operate the auxiliary drives.

The oil-mist method of lubrication has also increased the life of the anti-friction bearings on the tinning lines by preventing the seepage of acid fumes into the bearings. It is possible to prevent this contamination because, with the oil-mist method, the pressure within the bearings is kept at a ratio higher than the atmospheric pressure outside the bearings, and because clean air and oil pass into the bearings when air escapes through the seals.

Oil mist has also been successfully applied to furnace-fan bearings. It has been reported that a number of fans operating in very high temperatures required bearing replacements of three or four bearings per month. Since the application of oil mist, the changing of fan bearings is now, however, only rarely necessary.

These are only a few examples of the successful application of oil-mist lubrication in the steel industry. It has been similarly successful in the aluminium industry and in the chemical industry. Naturally, the oil-mist system can be applied to machines with various control accessories. For instance, heaters are available for the oil-mist units to keep the oil at a constant temperature and constant viscosity, entirely independent of all ambient temperatures surrounding the oil-mist units. Low-level devices may be applied to the units to give the operator sufficient warning when the level of the oil in the unit reaches a critical level. Gauges and signal devices may be applied both to the air line supplying the oil-mist unit to indicate any air failure and also to the mist line to indicate any failure in the line pressure.

A number of plants have done considerable work in machine-tool cutting and grinding with the oil-mist principle. However, it is considered that a great deal more work in both design and application needs to be done in order to improve this possibility.

There are only a few basic fundamentals which must be adhered to for the successful application of the oil-mist method of lubrication. The oil put into the oil-mist units must be kept clean, and care must be taken to ensure that the lubrication requirements of the bearings are within the capacity of the number of units applied. In addition, it is essential that the air lines supplying the mist lubricator and bearings be clean and free from moisture. In this connection, commercial oil-mist units are at present equipped with a water separator which will take out approximately 95% of the removable moisture from average factory air. However, air lines which are subjected to excessive water, scale, blow-by oil, carbon, etc., must use auxiliary filters and water separators.

Electrodynamic Standards for Vibration Pick-ups

(From *National Bureau of Standards Technical News Bulletin*, Vol. 41, No. 1, January 1957, pp. 11—15, 2 illustrations.)

A METHOD of standardizing electrodynamic calibrators for vibration pick-ups measuring displacement, velocity, or acceleration has been developed by the National Bureau of Standards, U.S.A., who have for some time been investigating various methods of calibrating pick-ups used for testing missiles, aircraft, ships, surface vehicles, and other types of equipment.

The electrodynamic calibrator for vibration pick-ups consists of a vibration exciter with a built-in vibration-measuring device. It produces mechanical vibrations by electrical means and simultaneously generates an electrical output which depends on the vibration. The calibrator is standardized by determining the relationship between its electrical output and the vibratory motion. Once an electrodynamic calibrator has been standardized, the procedure for calibrating a pick-up is straightforward. The pick-up is attached to the mounting table of the calibrator and is set in vibration at the desired amplitude and frequency. The amplitude can be accurately set as a result of the standardization procedure, and the frequency can be accurately measured, after which all that is necessary is to compare the voltage output of the calibrator with that of the pick-up.

The NBS method of standardizing the calibrator is based on an improved mathematical analysis of the problem—a more comprehensive reciprocity theory. This method avoids all direct measurement of the vibratory motion itself. It also has the advantage that it can be used above, as well as below, certain resonances in the calibrator. It is only necessary to measure frequencies, currents, and voltages in the calibrator when (1) several known masses are attached to the mounting table, and (2) when the calibrator is forced to vibrate by an external vibration exciter.

The result of the standardizing procedure is summed up in two quantities a and b which characterize the calibration factor of the calibrator. This factor then expresses the relation between the voltage output of the calibrator and the velocity of the mounting table. At higher frequencies, starting at about 900 cps, it is also necessary to know the mechanical impedance of the pick-up being calibrated. If not already known, this can easily be measured with the standardized calibrator. In calibrating a pick-up, both the magnitude and phase of the pick-up calibration factor can be determined, if desired.

Electrodynamic Calibrator

The electrodynamic calibrator for which the present method has been developed has three basic components, i.e., a driving coil (1), a mounting table (2), and a velocity-sensing coil (3), as shown in Fig. 1. These components are the principal moving parts. In the calibrator studied, these parts, joined by a shaft, form a single mechanical unit suspended from a frame by leaf springs (4). Also attached to the frame are a field magnet coil (5) near the driving coil and a permanent magnet (6) near the velocity-sensing coil. When alternating current is passed through the driving coil (1), its magnetic field interacts with that of the field coil (5), causing the moving parts to vibrate. At the same time, because of the motion relative to the permanent magnet (6), an alternating voltage is generated in the velocity-sensing coil (3).

In using the calibrator, the pick-up to be calibrated is attached to the mounting table (2) and its electrical output compared with that of the velocity-sensing coil (3). If this comparison is to be significant, however, the calibrator must be standardized. In other words, the relation between the electrical output of the velocity-sensing coil and the velocity of the table must be determined. The reciprocity method enables this relation to be calculated without requiring a detailed knowledge of the structure of the calibrator.

Reciprocity Theorem

Reciprocity theory was first applied to electrical circuits. As applied to the electrodynamic calibrator, the reciprocity theorem used states that the ratio of mechanical force at the mounting table to the driving-coil current, when the velocity of the table is zero (i.e., when it is held fixed), is equal in magnitude and opposite in phase to the ratio of the terminal voltage of the driving coil to the mounting-table velocity, when the driving-coil current is zero (i.e., when its circuit is open).

In its present form, the theory applies to an electrodynamic calibrator operating in any frequency or amplitude range in which it can be considered linear. Neither the internal structure nor the magnet assembly need be assumed rigid. It applies, for example, not only at low frequencies and at frequencies near certain resonances, but also at frequencies above axial resonance, when the driving coil and mounting table move in opposite directions. In addition, the extended theory takes account of electrical coupling between different parts of the same coil and between parts of different coils.

Standardization Procedure

Standardization of the calibrator determines the calibration factor F (the ratio of induced voltage in the velocity-sensing coil to the mounting-table velocity). This factor depends mainly on the characteristics of the calibrator itself, but it is also influenced at higher frequencies by the mechanical impedance of the pick-up attached to the table.

When a vibration pick-up is attached to the table and the driving coil is energized with alternating current at any particular frequency, the theory shows that

$$F = a + bY_p \quad .. \quad .. \quad .. \quad (1)$$

where Y_p is the mechanical impedance of the pick-up, and a and b are constants depending on the construction of the calibrator. All the symbols stand for complex numbers in the manner used in alternating-current theory. Thus, Y_p is the complex number that represents in magnitude and phase the ratio of the sinusoidal force applied to the pick-up to the resulting sinusoidal velocity. The aim

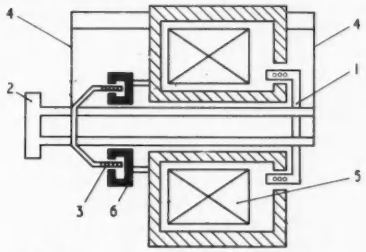


Fig. 1. **Schematic cross-section of electrodynamic calibrator for vibration pick-ups.**

of the standardization procedure is to determine the constants a and b. In computing their values, transfer-admittance and voltage-ratio measurements are made.

The transfer admittance G_w, which is the ratio of driving-coil current to velocity-sensing-coil voltage, is measured with each of several weights W, attached to the mounting table. The transfer admittance G_0, with no weight attached, is also measured. Then, for each weight attached to the table, the ratio $W/(G_w - G_0)$ is computed. This procedure is repeated at each frequency for which the values of a and b are desired. For each frequency, the real and imaginary parts of the computed ratio are plotted against W.

For voltage-ratio measurements, the mounting table of the calibrator being standardized is mechanically connected to that of another electrodynamic calibrator or other vibration exciter. The driving coil in the latter is energized and the ratio R of the open-circuit voltage generated in the velocity-sensing coil to the open-circuit voltage generated in the driving coil is measured. This is effected for each of the frequencies used in the transfer-admittance measurements. The voltage-ratio measurements are made, of course, on the calibrator being standardized.

Once these measurements have been made, the constants a and b are computed from the following equations:—

$$a = 0{\cdot}01711\sqrt{j\omega JR} \qquad b = 6{\cdot}601\ Q\sqrt{R/(j\omega J)} \quad (2)$$

where ω is the frequency in radians per second and j is the unit imaginary vector. The terms J and Q are the ordinate intercept and slope respectively of the plot of $W/(G_w - G_0)$ against W. This completes the standardization procedure, as the calibration factor can be computed from eq. (1) if the mechanical impedance of the pick-up to be attached to the mounting table is known.

The procedure can be illustrated by results obtained for a typical electrodynamic calibrator with a nominal 50-lb driving-force rating. Calibration factors for this instrument were computed from eq. (1) for pick-ups with mechanical impedances corresponding to weights W of 0, 0·5, and 1·0 lb (Y_p is then equal to $j\omega W/g$, where g is the acceleration due to gravity). The magnitude and phase angle of these calibration factors were plotted against frequency. The graph showed that, for this particular calibrator, the calibration factor is independent of frequency and pick-up weight up to 900 cps.

Calibration of Pick-ups

Ordinarily, before a pick-up can be calibrated, its mechanical impedance Y_p must be known, in order to use eq. (1) to obtain the calibration factor of the calibrator. If the value of Y_p of the pick-up is not known, it can be measured on an electrodynamic calibrator standardized by the reciprocity method. The theory shows that

$$Y_p = j\left(\frac{\omega}{386}\right)\left(\frac{J(G_p - G_0)}{1 - Q(G_p - G_0)}\right) \quad .. \quad (3)$$

where G_p is the transfer admittance (ratio of driving-coil current to velocity-sensing-coil voltage) with the pick-up attached to the table. The values of J, Q, and G_0 are known from the standardization of the calibrator.

One method of calibrating a pick-up is to energize the driving coil of the calibrator at the desired frequency and to drive the calibrator at the desired amplitude, as indicated by the voltage in the velocity-sensing coil. From this indicated voltage and the value of F obtained from eq. (1), the velocity of the mounting table can be found by simple division. The output of the pick-up corresponding to this velocity is then measured. The calibration factor of the pick-up is then the ratio of the output of the pick-up to the velocity of the mounting table.

A second method is to measure G_p and also the transfer admittance I/E_p, which is the ratio of the current I in the calibrator driving coil to the voltage output E_p of the pick-up. The calibration factor F_p of any accelerometer is then given by

$$F_p = \left(\frac{G_p}{I/E_p}\right)\left(\frac{gF}{j\omega}\right) \quad .. \quad .. \quad (4)$$

where $gF/j\omega$ is the acceleration calibration factor of the calibrator. The same method can be used to calibrate velocity- and displacement-type pick-ups, except that eq. (4) must be modified by replacing the acceleration calibration factor by the calibration factor F for velocity or for displacement $j\omega F$, respectively.

For example, in the case of a piezoelectric accelerometer, the transfer admittance G_p was measured with the device attached to the mounting table, and its mechanical impedance Y_p was found from eq. (3). The other quantities in eq. (3) were known from the calibrator standardization data. The calibration factor F could then be obtained from eq. (1). After measuring the transfer admittance I/E_p, the calibration factor of the pick-up F_p was computed from eq. (4).

The calibration of a variable-resistance accelerometer was performed in a similar manner. In determining the calibration factor for such a pick-up at frequencies other than those at which the calibrator was standardized, it was necessary to interpolate for a, b, R, and Q, and to compute J from eq. (2). This interpolation was used only at frequencies below 900 cps, in which range the interpolation leads to no appreciable error for the calibrator used.

It was found in the calibration of these pick-ups that the term bY_p in eq. (1) was negligibly small compared with the term a at frequencies up to 900 cps. In such cases, the procedure in both of the above methods is considerably simplified. Once the calibrator has been standardized, it is easy to predict the frequency range in which the term bY_p can be neglected.

Calibrator Resonances

Resonant conditions of the calibrator studied include resonance of the mounting table and coils as a rigid body on the guiding leaf springs, local resonance in the leaf springs, transverse resonance of the shaft connecting the mounting table and driving coil, longitudinal, resonance the same shaft, and local resonance in the mounting table or coils.

Longitudinal resonance of the driving coil relative to the velocity-sensing coil reduces the accuracy of the reciprocity calibration over a small frequency range. This is because R at resonance is very small and therefore difficult to measure, and because at resonance the large amplitude of the driving coil may exceed its linear range.

Transverse motion from any source invalidates the reciprocity method, because the latter requires the mounting table to have uniaxial motion. Usually, transverse motion occurs at frequencies for which the leaf springs or shaft are in transverse resonance. The leaf springs of some calibrators can be detuned, thus changing the frequency at which resonance occurs, by repositioning small weights attached to them. When

(*Concluded on page 85*)

Sliding Contacts for Heavy Currents and High Sliding Speeds

By P. KLAUDY. (From *Maschinenbau und Wärmewirtschaft*, Vol. 11, Nos. 10/11, October/November, 1956, pp. 315-326, 16 illustrations.)

In our issue of November 1955, we published an article on "The Properties of Sliding Contacts" by Professor Klaudy. In the following article, the author reports on further investigations in the field of sliding contacts for heavy currents up to 100,000 A and sliding speeds up to 60 m/sec.

WHEN two cylindrical metal contact members are pressed together by means of an axial load, contact occurs only over comparatively small local contact areas ("Hertz contact areas") of the order of 0·01 mm², owing to the residual surface roughness, and is obtained even with clean end faces having a high surface finish. When a voltage is applied to the contact members, the flow of current, instead of forming an ideal pattern of parallel lines, concentrates at the contact points. These create a "constriction resistance" which is greater than that corresponding to the full cross-section of the contact members. It has been found that the constriction resistance R_K is dependent only on mechanical load and temperature, under otherwise similar conditions (shape of contacts, hardness, etc.), decreasing with an increase in load, and increasing with temperature.

The value of R_K can be calculated for simple cases; for instance, for two infinitely large conductive media M_1 and M_2, separated by a thin insulating layer with only a circular area or "opening" of radius r over which contact occurs, the constriction resistance is

$$R_K = (\rho_1/4r) + (\rho_2/4r) \quad .. \quad .. \quad .. \quad (1)$$

where ρ_1 and ρ_2 = resistivities [ohm-m] of the two media. It may be noted that about $\frac{1}{8}$ of the total voltage drop due to R_K occurs within an area of radius $2r$ surrounding the centre of contact.

Numerical evaluation of eq. (1) shows that the constriction resistance is extremely low for good conductive materials. For copper, assuming $r = 1$ mm, with $\rho_1 = \rho_2 = 1{\cdot}75 \times 10^{-8}$ ohm-m, the result is only $R_K \cong 10^{-5}$ohm.

Although pure metal surfaces can be obtained in vacuum by heating to incandescent temperatures, they tarnish very rapidly in contact with air, i.e., gas atoms, molecules, or ions are adsorbed and bonded to the metal surface, in many cases after conversion into chemical compounds (oxides, hydroxides, carbonates, etc.). The foreign layers are then built up from monomolecular films to thicknesses of hundreds or thousands of angstroms by metal ions diffusing through the existing layers or attracted by electrical forces to the outer surfaces, where they react with gas ions from the atmosphere, to form further layers of a thickness depending on the nature of the metal. Thick foreign layers have the properties of semiconductors or electrolytes. The total contact resistance R_C is then not only that of the constriction resistance R_K but also includes the resistance of the foreign layer, so that

$$R_C = R_K + (\rho h/a) \quad .. \quad .. \quad (2)$$

where ρ = resistivity of foreign layer (of the order of 10^4 ohm-m), a = local contact area, and h = maximum gap between contact members at the position surrounding the contact area.

Thus, the term $\rho h/a$ for the resistance of foreign layers reaches very high values, of the order of 10,000 ohms. Experimental evidence, however, shows that such resistances are not met with in practice. It has been found that the effective current transmission through tarnished surfaces is mainly attributable to three causes, i.e., (1) the tunnel effect, (2) crushing and fusion (fritting) of foreign layers, and (3) bridges of conductive material occurring in the foreign layers, owing to high electric fields.

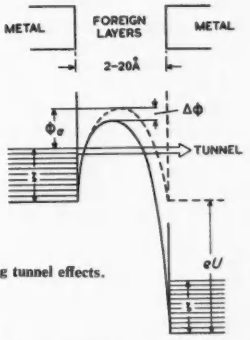


Fig. 1. Conditions producing tunnel effects.

According to electron theory, the free electrons in metal lattices are in a state of thermal motion, but their velocity distribution is independent of temperature. Those with the highest kinetic energy ζ are still not at a sufficiently high energy level to overcome the retaining forces of the metal lattice near its boundary. To do this, they would have to perform the work Φ_a required to emerge from their uppermost level ζ to the surface of the metal. This work is reduced, however, by an external electric field (in Fig. 1, the dotted parabola is lowered by $\Delta\Phi$ to a similar curve shown as a solid line), owing to an applied potential U between the contact members, causing a reduction eU of the energy required (e = electron charge). By properties associated with wave mechanics, some of the highest-energy electrons can then pass or "tunnel" their way through the remaining potential barrier and produce a "tunnel current", which can be calculated. This current, as indicated by theory and confirmed experimentally, is independent of temperature, i.e., its temperature coefficient is $\alpha = 0$. For thin foreign layers, of thicknesses between 2 and 30Å, the tunnel resistance R_T, i.e., the ratio of contact voltage to tunnel current, is of the order of 10^{-9} to 10^{-12} ohms per m² of tunnel-contact area; in other words, it is several powers of ten lower than that of foreign layers as calculated by eq. (2). In such cases, the total contact resistance R_C consists only of the constriction resistance R_K and the tunnel resistance R_T, so that

$$R_C = R_K + R_T \quad .. \quad .. \quad .. \quad (3)$$

The tunnel contact areas are smaller than the Hertz contact areas and are situated within these contact positions.

With thick foreign layers, the tunnel current is practically reduced to zero. Conduction of current can then occur only if the mechanical loads are sufficiently high to squeeze or crush the foreign layers. Conductive material is forced through the gaps or cracks thus produced, and in this way bridges of conductive material

are formed between the two contact members. The "bridge resistance" R_B is an ohmic resistance which is added to the constriction resistance R_K, so that the total contact resistance in this case is

$$R_C = R_K + R_B \quad .. \quad .. \quad (4)$$

Even at low mechanical loads, however, conduction can occur through thick foreign layers if the electric field strength has a sufficiently high value. With fields of the order of 10^8 V/m the electrical forces lead to local increases in conductivity and current density, with the formation of metallic compounds and partial fusion (fritting). The foreign layer is thus punctured by bridges of conductive material. The total contact resistance is again calculated by means of eq. (4), which is associated with a positive temperature coefficient ($\alpha > 0$).

Sliding Contacts

Whereas terminal connections, switches, cut-outs, etc., are nearly always designed with sufficiently high contact loads to ensure conduction of current either by a tunnel effect or by a squeezing action, high loads cannot be applied to the same extent to sliding contacts, since they would result in high friction losses, danger of fracture, and a high rate of wear. Also, with sliding contacts the problem of rough running, mechanical impact, and resonant vibrations must be considered.

For sliding contacts, therefore, combinations of materials with good sliding properties are used, e.g., carbon brushes running on copper or bronze slip-rings. The carbon, because of the laminar structure of graphite, is easily displaced in certain lattice directions and fills in any irregularities in the slip-ring surface. Good sliding action is also due to the low frictional coefficient of graphite, the formation, through rubbing action, of smooth layers of oxide or carbonate (about 20 to 30Å in thickness) adhering to the slip-ring surface, and the adsorption of thin films of water. Normal brush loads are between $0{\cdot}2 \times 10^4$ and $1{\cdot}0 \times 10^4$ kg/m^2. Conduction is predominantly due to fritting. In view of the high values of field strength occurring in the oxide layers, the voltage drop of the fritting type of contact of the sliding members is in the range of 0·2 to 1·5 V.

The power loss through heat dissipation in sliding contacts can be calculated as

$$P_E = 2V_C I_C \times 10^{-3} \text{ [kW]} \ldots \quad .. \quad (5)$$

where V_C = fritting contact voltage [V], and I_C = contact current [A]. Permissible values of current density S are therefore limited to values between 5×10^4 and 25×10^4 A/m^2. The power loss P_M due to mechanical friction can be evaluated as

$$P_M = 2 \times 10^{-2} \mu p I_C v / S \text{ [kW]} \quad .. \quad (6)$$

where μ = coefficient of friction, p = brush pressure [kg/m^2], and v = sliding velocity [m/sec]. As an example, it may be noted that, in the case of an arrangement transmitting 10,000 A at a sliding velocity of 60 m/sec with metal/graphite brushes, the total power losses of the brushgear were evaluated as $P_E + P_M = 20 + 50 = 70$ kW.

Considerable research is therefore being concentrated on improving brush-type sliding contacts; with new designs of brushes, slip-rings, and cooling arrangements, definite improvements can be expected, but the basic difficulty with solid sliding contacts remains. To limit mechanical losses, contact loads have to be kept small; low contact loads, however, result in current conduction, mainly through fritting, and, since this is associated with an appreciable voltage drop, high electrical losses are obtained.

Liquid Contacts

Another method of obtaining sliding contact is to interpose a conductive liquid between the contact members. This has the advantages of reduced rubbing losses, improved cooling possibilities by continuous replacement of contact liquid, and considerable reduction or even elimination of contact resistance (with untarnished surfaces), since the liquid is in contact with all points of the macroscopic surface irregularities of the contact members.

Although electrical contacts containing mercury have occasionally been employed since the earliest days of electrical technology (Faraday's rotational experiments, Barlow's wheel, etc.) and are nowadays used mainly in mercury switches, information on their behaviour is nowhere to be found in technical literature, and it was therefore necessary to carry out various investigations to obtain basic data.

The first tests were made on a stationary arrangement consisting of a thick-walled tube section of perspex containing the two contact members, with the mercury introduced between them. This tube was fitted in the circular grooves of two metal supports and sealed by means of sealing rings. The upper support was provided with a central passage passing through the upper contact member, to enable the mercury to be introduced after preliminary evacuation. The upper section of this passage was then filled with oil at pressures between zero and 300 atm, adjustable by means of a hand pump. The lower support incorporated a plunger, adjustable by means of a screw, which made it possible to vary the gap.

Contact resistance was evaluated from current and voltage measurements, the voltage tappings being obtained by means of steel or platinum probes passing through the perspex cylinder into the two contact members, the probes being connected to a light-spot galvanometer. Various distances between the probes were tried until the most suitable positions were found. All necessary precautions were observed (polishing, cleaning, and drying of contact members; distillation and chemical cleaning of mercury; measurements at constant temperature with reversed current polarity, taking mean values in order to eliminate local thermoelectric effects; annealing of probe points in vacuum, etc.).

It was expected that liquid pressure should have no influence on the conduction of current and this was amply confirmed by tests with various materials for the contact members. In contrast, solid contact arrangements show a hyperbolic drop in resistance with increasing pressure. The contact members, 10 mm thick, were made of a steel specially selected because it does not form a mercury amalgam. Molybdenum contact members were also tested. Steel, unlike molybdenum, shows corrosion phenomena in moist air; these are probably due to local element effects, as indicated by Tuchscheerer, and may cause sudden irregular variations in contact resistance.

In all cases, the basic (ohmic) resistance of the contact members was calculated from dimensions and resistivity values, and subtracted from the measured resistance in order to obtain the effective contact resistance. In tests serving to establish that pressure had no influence, values of the order of 20×10^{-5} to 30×10^{-5} ohms were obtained, these being attributed to the bridge resistance of the foreign layers. Despite cleaning, these layers either

remained on the surfaces or were produced in the period between cleaning and measurement.

To obtain a further reduction in contact resistance, the requirement was to avoid the formation of foreign layers on the contact-member surfaces. By turning these members in a nitrogen atmosphere and inserting them into the test apparatus as quickly as possible (within 20 sec), reduced values of resistance were achieved. However, completely successful results were finally obtained by a special technique involving milling under mercury.* With this technique it was found possible to eliminate contact resistance entirely, the only remaining resistance being the basic (ohmic) resistance of the contact members themselves and the mercury, as determined by their resistivities and the dimensions used.

This milling and testing procedure was used to investigate various materials employed as contact members. It was found that, as long as the clean surfaces remained under mercury, the contact resistance remained zero, even after some months. Foreign layers, indicated by definite values of contact resistance, appeared only when the contact members were exposed to environmental influences. When the members were exposed to air for a few seconds only and then immersed in the mercury, it was found that the thin foreign layers disappeared gradually after immersion, whereas thicker layers caused by longer exposures to air remained after subsequent immersion.

Typical results of exposure tests are as follows: Steel, in air at 20°C and 60% relative humidity, showed contact resistances, of 0, 4, 7·5, and 114 micro-ohms, after respective exposure times of 0 sec, 1 min, 30 min, and 15 hr. For molybdenum, the corresponding values were 0, 3·2, 4·9, and 9·8 micro-ohms, the latter value being obtained after 14 instead of 15 hr. In air at 20°C and 100% humidity, steel exposed for 5 hr gave a value of 1000 micro-ohms, whereas molybdenum after 29 hr gave only 14·8 micro-ohms. Steel exposed for 15 min in hydrogen at 1 atm had a resistance of 8 micro-ohms and for the same period in air at 1·3 micro-bars a resistance of 2·5 micro-ohms. Further tests were made in nitrogen, hydrogen, and oxygen, at various pressures and temperatures. Other materials investigated included nickel, tungsten, chromium, platinum, and carbon. Nickel and tungsten, in particular, showed less susceptibility to environmental influences, but somewhat higher contact resistances, than molybdenum. On carbon no traces of foreign layers were found, but the resistivity of carbon is, of course, higher.

To avoid machining difficulties and to reduce costs, the materials can be applied in the form of coatings to the contact members, which may be of copper, wrought iron, or other materials. Sprayed-on coatings were found to be unsuitable, owing to slag residues and the lack of purely metallic bonding at the contact positions; sputtering and electrolytic deposition were only partially satisfactory, because of gas residues in the layers or between the layers and contact members. After many attempts, the author and his collaborators succeeded in applying coatings of tungsten and (for the first time) molybdenum on steel, by means of electric welding.

The effect of foreign layers on the contact members of liquid contacts was found to be very much less than with solid contacts. Plates with thick layers of rust, with a cross-sectional area of $5 \times 20 = 100\,cm^2$, were placed in a mercury-filled trough, and measurements gave relatively low values of contact resistance, e.g., 10^{-3} ohm at 10μV.

* *The Engineers' Digest*, November 1955, p. 518.

The question as to the type of current conduction occurring in such conditions was then considered. Crushing or squeezing effects could not occur with liquid contacts, but tunnel or fritting effects were conceivable, though the latter could hardly be expected to have occurred at the very low voltages used, which resulted in low field strengths. It seemed more probable that the low resistance was due to conduction of the foreign layers, which in these tests were entirely covered with mercury, whereas in solid contacts the layers are in contact with each other in small localized areas only.

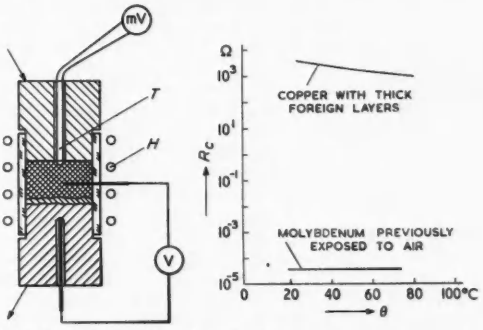


Fig. 2. Temperature dependence of contact resistance in liquid contacts.

To clarify this point, further tests were carried out on contact members with foreign layers on their surfaces, using a cylindrical test apparatus (Fig. 2) with a glass tube, similar to that previously employed, but provided with an external heating coil H, in order to determine the temperature coefficients. The temperature θ of the contact arrangement was measured by means of a thermocouple T inserted in the upper passage used for filling. It was found, for instance, that copper with a thick foreign layer had a definitely negative temperature coefficient (characteristic of conduction in semiconductors), in contrast with molybdenum exposed to air, which had a constant contact resistance R_C, independent of temperature (Fig. 2).

Similar contact members of 10 mm diameter were also employed, to determine the variation in contact resistance

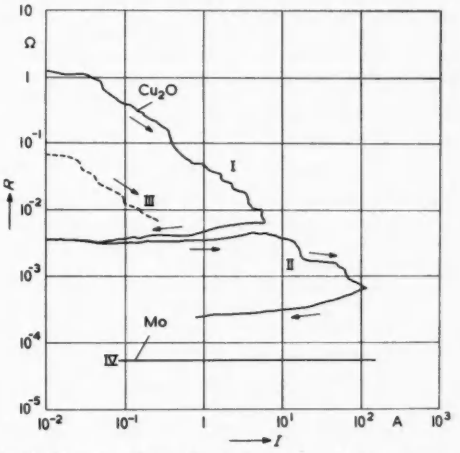


Fig. 3. Contact resistance of contact members with mercury-covered foreign layers.

with different current intensities. The current was varied from zero to 150 A, and the results obtained are shown in Fig. 3. For the copper contact members, the contact resistance was at first fairly constant (curve I) at very low currents and contact voltages. This was evidently the ohmic resistance due to the conductivity of the foreign layers. Beyond a certain value, the resistance decreased suddenly (at 0·05 A), because the field strengths were sufficient to cause fritting, i.e., fusion of conducting material through the outer layers. After reaching a value of 5 A, the current was reduced to zero and the resistance remained at its lowest value, indicating that the conducting bridges were not altered by reducing the current after they had been formed. A subsequent increase in current (curve II) showed that the contact resistance hardly changed until the current exceeded its previous maximum, when a further falling off in resistance was recorded. In this range, the increased field strengths in the foreign layers caused further fritting phenomena. The current was then again reduced to zero and the contact resistance retained its lowest value. After this, the mercury was removed from the apparatus and the copper contact members were exposed to air for several hours. A further test gave the results indicated by curve III. These showed that the exposure to air caused a partial "healing" of the bridges previously obtained through fritting, by the formation of new foreign layers which were subsequently "punctured" by a current somewhat lower than that initially required.

The behaviour of the molybdenum contacts which had been exposed to air was quite different, as shown in Fig. 2 and by curve IV in Fig 3. Their contact resistance, considerably lower than that of copper with thick foreign layers, was independent of temperature and current or applied voltage; these features indicated that the conduction of tarnished molybdenum surfaces was due to the tunnel effect through relatively thin films.

The results and conclusions derived from the above tests were also confirmed in many subsequent investigations.

Mechanical Properties of Liquid Sliding Contacts

In the arrangement employed for testing liquid-type sliding contacts (Fig. 4), mercury was fed into a contact gap through a channel in a stationary contact member, in such a way that its flow velocity was negligible compared with the peripheral velocity v of the rotating contact member. For simplicity, it can be assumed that there is a linear velocity distribution in the mercury, with zero velocity at the boundary of the stationary contact member and the velocity v near the rotating member. The power loss due to viscous friction is then

$$P_F = \eta A v^2/s \ \text{[W]} \quad .. \quad .. \quad (7)$$

where η = viscosity of mercury [newton-sec/m^2], A = contact-member area covered by mercury [m^2], v = peripheral velocity [m/sec] of the rotating contact member (slip-ring), and s = height of gap [m].

In addition, there is a mechanical loss due to the fact that the mercury leaving the gap has a kinetic energy which cannot generally be recovered. This power loss (assuming a linear velocity distribution) can be calculated as

$$P_K = \gamma v^3 b s/16 \ \text{[W]} \quad .. \quad .. \quad (8)$$

where γ = density of mercury [kg/m^3] and b = width [m] of contact member covered by mercury.

There are also eddy losses due to the non-laminar

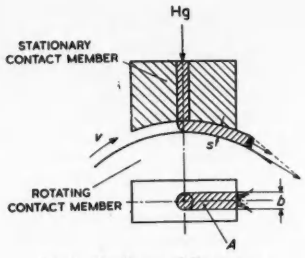


Fig. 4. Liquid-type sliding contacts.

flow of the liquid and the partial breaking up of the jet of mercury on the rotating contact member, as well as capillary effects. These extremely complicated conditions, observed in tests using transparent contact members, are difficult to express mathematically.

In practice, the losses obtained are generally lower than those indicated by eqs. (7) and (8). This is probably due to the low adhesion of mercury to metals, and to its high density and low viscosity. Suitable s-values for gap height are between 0·2 and 0·4 mm. With these small values, only a very thin layer of mercury is entrained, the velocity distribution is non-linear, and the kinetic losses are much lower than those expected from eq. (8). From test results, it appears that P_K is proportional to v^n, where n is an exponent between 2 and 3.

Design Considerations for Liquid Sliding Contacts

In designing liquid-type sliding contacts, the endeavour will, of course, be to make the total losses as small as possible. In addition to the mechanical losses (eq. (7) and (8)), the electrical power losses have to be considered. These can be evaluated as

$$P_{el} = I^2 R_{tot} \ \text{[W]} \quad .. \quad .. \quad (9)$$

where

$$R_{tot} = \frac{0{\cdot}9\rho}{\sqrt{A}} + \frac{c}{A}\,10^{-10} + \frac{s}{A}\,10^{-6} \ \text{[ohms]} \quad .. \quad (10)$$

where ρ = resistivity of contact-member material [ohm-m], A = area of contact [m^2] covered by mercury before breaking away from the upper contact surface, c = a value between 0·01 and 10·0, depending on the thickness of the foreign layers on the contact members, and s = height of gap [m]. It may be noted that, in eq. (10), the terms on the right-hand side represent the constriction resistance of the contact members, the tunnel resistance of the foreign layers on the contact members, and the ohmic resistance of the mercury, respectively.

Consideration of eqs. (7) to (10) shows that the requirements for low power losses are not all easy to fulfil simultaneously. For instance, for low electrical losses, large A-values and small s-values are desirable, but this would result in increased frictional and kinetic losses, although, as regards the latter, the value of b can be modified to some extent. For the design of low-loss liquid-type sliding contacts, it is considered that

(1) The contact area covered by the conducting liquid should be as small as possible (of the order of a few mm^2).

(2) The contact members should be of highly conductive materials which are not attacked or corroded by the contact liquid; they should have as little tendency as possible to form foreign layers and should resist high centrifugal forces (on the rotating contact member).

(3) The gap s should be comparatively small (values between 0·2 and 0·4 mm are sufficient to cope with thermal expansion while maintaining low flow volumes for the contact liquid).

(4) The contact liquid should have good conducting properties not affected by air, low viscosity, and a freezing point which is as low as possible.

In practice, mercury is the liquid mainly used. Gallium and Wood's metal, and other alloys, are eliminated either because of cost or because of their high melting point or sensitivity to air. For the contact members, molybdenum, tungsten, and nickel are the main materials to be considered; in some designs, they can be employed in the form of coatings on other highly conductive materials.

In laboratory tests, a liquid-type sliding contact was used to transmit a current of 1000 A with a sliding speed of 150 m/sec; the total contact resistance was 10^{-3} ohm and the total power loss 700 W. In comparison with other known types of sliding contacts, this represents considerable progress. Investigations on practical applications are now being carried out.

RUSSIA

A New Machine for the Form-Grinding of Blade Profiles

By I. I. Stepanov. (From *Stanki i Instrument*, No. 9, September 1956, pp. 5–7, 6 illustrations.)

This article reports the first stage in the development of a machine for the mass production of complex curved surfaces, mainly of the convex type, by form-grinding. The machine embodies an unusual principle, designated " conjugate-profile grinding ", in which grinding is effected by a wheel profiled both radially and across its width. General observations of the method and its limitations are given, together with design solutions to a number of problems encountered and indications of the initial difficulties experienced with the first machine operating on the new principle.

Designed for the form-grinding of complex curved surfaces, such as the convex side of gas-turbine and compressor blades, the machine described in this article can machine two or four blades by the method of plunge-grinding, using a wide grinding wheel profiled in two dimensions, the width of the wheel being equal to the length of the blade. The machine can also be used for grinding the convex side of blade roots, while the number of components which can be ground simultaneously is partly determined by their shape.

The machine operates semi-automatically. After loading the component on a mandrel, mounted between the centres of the work-spindle, the operator turns a handle, causing the drive to be switched on and the grinding-wheel spindlestock to advance rapidly towards the component. This is followed by feed motion of the grinding wheel against the root shoulder, after which the rough- and finish-grinding plunge feeds take place. The cycle is continued by the withdrawal and rapid removal of the grinding-wheel spindlestock, whereupon rotation of the work-spindle and the supply of coolant cease. The operator then removes the finished components from the mandrel, replaces them with the next batch, and the cycle is repeated.

The rate of longitudinal and plunge feeds, both in the rough- and finish-grinding stages, and also the duration of running out, are controlled hydraulically. The work-spindle is driven positively by a chain transmission from the grinding-wheel spindle, with a gear ratio of 1:2 or 1:4. During one revolution of the grinding wheel, the work-spindle rotates through an angle equal to that between two components, i.e., it advances by one component.

When wear of the grinding wheel is excessive, the operator engages a wheel-truing mechanism, which is controlled by a special two-dimensional master cam, rotating in synchronism with the grinding wheel. The grinding wheel, 700 to 1100 mm in diameter and of a width equal to the length of the blade, is shaped during truing by means of a carbide roller mounted at the end of a lever attached to the shaft of the truing device on the grinding-machine table. The other end of the shaft carries another lever, on which is mounted a rotating follower, pressed against the master cam during truing operations by a spring. The master cam, which is rotated by the work-spindle through a gear train, has a profile corresponding with that of a single blade. When the table and the truing device move and the master cam is rotated, the follower pressing against it transmits the necessary displacement to the truing roller, both across the width of the wheel and in all radial cross-sections.

Owing to the difference between the diameters of the wheel and the component, a relative sliding of their profiles takes place at their line of contact. The mean sliding speed is approximately equal to the usual speed in cylindrical grinding, because the mean diameter of the grinding wheel is between 70 and 100 times greater than that of the component being ground, and the angular velocity of the grinding wheel is either 2 or 4 times greater than that of the work-spindle, the nominal grinding speed being 30 m/sec.

The Kinematics of Profile Formation

The instantaneous centre of the grinding wheel and the component profiles when both are rotating in the same direction (thus exemplifying an opposed grinding motion, as embodied in the new machine) can be found as follows :— In Fig. 1, O_1 and O_2 are the respective

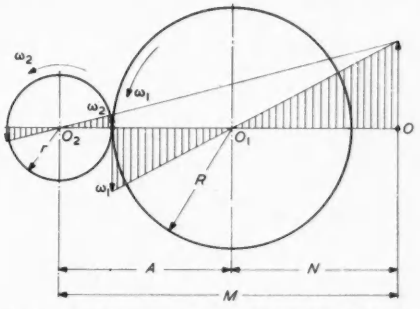


Fig. 1. Location of instantaneous centre with grinding-wheel and work-spindles rotating in the same direction.

centres of rotation of the grinding wheel and work-spindle, O being the common instantaneous centre, ω_1 the angular velocity of the grinding wheel, and ω_2 the angular velocity of the work-spindle.

As is well known, the contact points associated with O_1 and O_2 through O have instantaneous velocities which are equal in magnitude and direction, so that, if N is the distance between the centre of rotation of the grinding-wheel spindle and the instantaneous centre O, and A is the centre distance between the grinding-wheel and work-spindles,

$$\omega_2(N + A) = \omega_1 N \quad .. \quad .. \quad (1)$$

As stated above, the angular velocity of the grinding wheel is either two or four times that of the work-spindle, so that either $\omega_2 = \omega_1/2$ or $\omega_2 = \omega_1/4$.

If $\omega_2 = \omega_1/2$, then, from eq. (1), $N = A$, so that

$$M = (N + A) = 2A = 2(R + r) \quad .. \quad (2)$$

where M is the distance between the centre of rotation of the work-spindle and the instantaneous centre O, R is the instantaneous radius of the grinding-wheel profile, and r is the instantaneous radius of the component profile.

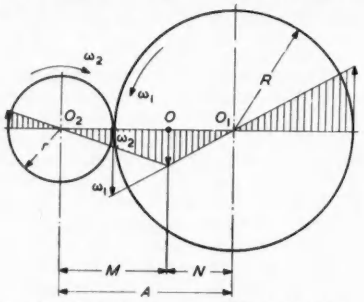


Fig. 2. Location of instantaneous centre with grinding-wheel and work-spindles rotating in opposite directions.

In this case, it can be seen from the position of the instantaneous centre that internal meshing of the profiles takes place.

If $\omega_2 = \omega_1/4$, it can be shown that $M = 1\frac{1}{3}A$. Here again, internal meshing takes place, but the instantaneous centre is located nearer to the centres of rotation. This situation creates more favourable conditions for profile generation.

If the grinding-wheel and work-spindles rotate in opposite directions (Fig. 2), the instantaneous centre will be located between O_1 and O_2. The distance between the centre of the work-spindle and the instantaneous centre O can be found from the equation

$$\omega_2 (A - N) = \omega_1 N \quad .. \quad .. \quad (3)$$

With $\omega_2 = \omega_1/2$, then $A = N$, so that

$$M = A - N = 2A/3 \quad = 2(R + r)/3 \quad .. \quad .. \quad (4)$$

In this case, external meshing of the profiles takes place.

Once the point of contact of the two conjugate profiles and the instantaneous centre are known, the relative velocity at the point of contact, i.e., the slip velocity, can be found, from which the required speed of the grinding spindle can be determined.

Theoretical investigations have established that external meshing is more favourable for the formation of the profile and for the grinding process than internal meshing, as adopted in the new machine. Calculations have shown that, with the grinding-wheel and work-spindles rotating in opposite directions, the fluctuations in relative grinding speed arising from variations in curvature are reduced, particularly along flat sections of the profile. As a result, grinding conditions are improved and non-uniformity of grinding-wheel wear is reduced. Moreover, in the case of external meshing, no undercutting of the profile takes place during generation, particularly when grinding wide and flat sections of the profile. Nevertheless, it was decided, at this stage of development of the new machine, to forgo the advantages of external meshing, in order to reduce the effect of backlash throughout the feed mechanism between the grinding-wheel and work-spindles.

Braking Mechanism for Elimination of Backlash

When grinding curved surfaces by the conjugate-profile principle, velocity varies with the contact point between the grinding wheel and the work, owing to the varying curvature of the ground surface. As a result, grinding force changes during each revolution. In transmitting rotation from the grinding-wheel spindle to the work-spindle, a special braking mechanism has been provided, to ensure reduced backlash in the gear train. The advantage of this mechanism over other designs of brake lies in its low dissipation of energy and hence the elimination of heating up of the mechanism.

The arrangement of the gear brake, which is shown in Fig. 3, is as follows :— The gear-wheel (1), which takes part in the transmission of the drive from the work-spindle to the master cam, meshes with a pinion (2) mounted on and keyed to an intermediate shaft (3). The braking gear-wheel (4), freely mounted on a spigot of gear-wheel (1), has the same module and diameter as this gear-wheel but has one tooth less. To ensure meshing with pinion (2), the teeth of gear (4) are made suitably thinner. The braking gear-wheel (4) has ground faces and is sandwiched and compressed between the face of wheel (1) and a ring (5) by means of ten springs (6) equally distributed around the ring and supported by the flange (7) attached to gear-wheel (1).

During rotation of wheel (1), one of its teeth presses against a tooth of pinion (2) in a certain direction. The braking gear-wheel (4) has a transmission ratio corresponding to the difference of one tooth, so that, during one revolution of wheel (1), wheel (4) will rotate in relation to wheel (1) with an advance of one tooth per revolution. In this way a constant reverse torque is produced, providing a braking effort by virtue of the pressure against

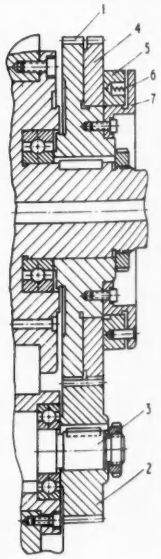


Fig. 3. Section through braking mechanism for elimination of backlash.

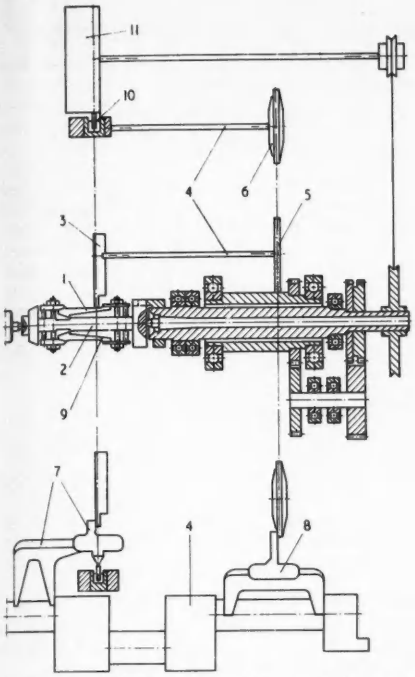


Fig. 4. Diagram showing the method used for profiling the master cam and truing the grinding wheel.

the tooth of pinion (2) in a direction opposite to that exerted by wheel (1). This counter-torque exerted by wheel (4) provides a wedge effect in the gear teeth and produces a transmission without backlash.

The grinding-wheel spindle runs in multi-pad bearings and, at its pulley end, is extended for the attachment of the chain transmission to the work-spindle. To compensate for the load on the grinding-wheel spindle occasioned by the weight of this transmission, a special device is provided. Since the grinding wheel itself is not round, ordinary compensation for balance is insufficient, and points for attaching additional balancing masses are provided.

Profiling the Master Cam

The master cam guiding the profiling of the grinding wheel is itself ground on the machine with the aid of a special fixture and truing device by the method of reverse copying from a master blade mounted on a mandrel in the work-spindle (Fig. 4). During grinding of the master cam, a follower (3) presses against the master blade (1) clamped on the mandrel (2). This follower has a working edge of 2mm (corresponding to the thickness of the truing roller (10)) and a profile corresponding to the mean radius of the grinding wheel (11).

During rotation of the mandrel (2), the follower (3) is pressed against the curved surface of the blade and receives a reciprocating linear motion which is transmitted through a shaft (4) to a crosshead carrying a grinding wheel (5), by means of which the profile of the master cam is ground by the displacement of the table of the truing device, together with the follower and the grinding wheel, along the blade.

The grinding wheel (5) for form-grinding the master cam has the same diameter as the roller (6), which follows the master cam during truing. The follower (3) and the grinding-wheel truing device are set by means of templates (7) and (8). During one revolution, the mandrel (2), carrying the master blade, turns by one half or one quarter of a revolution, thereby ensuring that the master-cam profile corresponds with the profile of one blade. During subsequent rotation of the mandrel, the follower is pressed against pads of much greater thickness than the blade, causing the grinding wheel (5) to withdraw from the master cam. When the mandrel has made one complete revolution, the follower is again in contact with the master blade, and grinding of the master cam is resumed. In this way, production of the master cam from only one master blade is ensured.

After profiling the master cam, the roller (6) is set up by means of templates (7) and (8), this roller being identical in size and shape with the master-cam grinding wheel (5). At the same time, the follower (3) is replaced by the tungsten carbide truing roller (10).

The master blade used for the profile grinding of the master cam has a corrected profile intended to take account of peculiarities in the design of the truing device and of the calculated and empirical distortions occurring with this method of grinding.

Conclusions

It is claimed that, by reason of its rate of production and its modest demands on the skill of the operator, this grinding machine shows great merit. Its main drawback, however, is that of limited quality of finish and accuracy, necessitating dimensional polishing, particularly where the profile has flat sections. Along such sections the highest relative or slip velocity takes place, and consequently the abrasive grains of the grinding wheel are most heavily loaded in such regions, thus obliging frequent wheel truing.

Insofar as size is concerned, the only limitation is that of producing wide grinding wheels of large enough diameter. The use of stacked wheels to increase the total width is not altogether satisfactory, as they leave a trace on the component caused by the joint line.

With regard to other difficulties arising from this method of grinding, technical and design analysis suggests that most of these can be eliminated. This refers especially to the question of finish and accuracy over flat portions of the profile, and it is intended to produce non-uniform motion of the work-spindle, to slow down motion whilst the grinding wheel is in contact with flat or low-curvature portions of the blade surface.

A wider investigation of the conjugate-profile method of grinding shows that its practical and economic possibilities are real only when a number of components can be mounted on the same work-spindle, so that several repetitions of the conjugate grinding-wheel profile are distributed around its circumference. In other words, this method can be recommended only when the ratio of grinding-spindle and work-spindle speeds is 1:2 or higher, but not when a 1:1 ratio is dictated by the geometry of the component, as would be the case, for example, when grinding camshafts. In this instance, it is stated that sections with a large radius of curvature would be ground at so high an effective grinding speed, compared with average speeds, that wear of the grinding wheel and the resulting loss of finish and accuracy would not be brought within economic limits. Furthermore, it is clear that this concept of machining is essentially adaptable economically only to mass production.

U.S.A.

The Infiltration of Cermets for Improved Toughness

By C. G. GOETZEL and J. B. ADAMEC. (From *Metal Progress*, Vol. 70, No. 6, December 1956 pp. 101–106, 7 illustrations.)

As indicated in this article, composite structures produced by impregnating porous titanium carbide compacts wtih a metallic binder have better impact strength and oxidation resistance than conventional cermets.

THE long search for high-temperature materials for turbine components to operate beyond the temperature limit of the alloyed metals has led to a group of composite materials called cermets, based on structures incorporating ceramics, intermetallic compounds, and the so-called "hard metals". These composite materials usually consist of a hard refractory phase bonded by a more ductile metallic phase. The combinations studied have included such a variety of systems as molybdenum disilicide and nickel, aluminium oxide and chromium, zirconium boride and nickel, and titanium carbide bonded with nickel, cobalt, or iron-base alloys. The material which seems most promising at present is based on titanium carbide as the refractory phase and various nickel alloys as the binder metal.

Although such cermets have been produced over a wide range of binder-to-carbide ratios, the best combination of physical properties is obtained when the composition contains approximately 60% by volume of titanium carbide. Successful fabrication techniques include sintering the components above the melting temperature of the binder, hot-pressing at these temperatures, or infiltration of a porous carbide skeleton with the molten binder.

Materials produced by all three methods have met with varying degrees of success, but their lack of toughness has so far hindered their acceptance as structural materials for gas-turbine components in jet engines. In view of the persistent shortcomings of present cermet products, renewed attention is now being given to the use of so-called "graded" cermet products, in which the metal content in the cermet component varies along one or more directions, so that the physical properties of any given section can accommodate the imposed mechanical and thermal conditions. Such a gradation could provide ductility and toughness in areas which require these properties. The same areas would not be subjected to a combination of high temperature and high stress, so that either high-alloy cermets or pure alloys could be used. The problem, therefore, is primarily one of selecting appropriate materials and developing a fabrication technique to permit cermets of varying alloy content up to pure alloys to be combined into one component.

It has now been established that the infiltration technique can produce certain types of graded titanium carbide products which possess the desired properties in each critical area. Thus, it should be possible to overcome the shortcomings in ductility and impact resistance now encountered in the homogeneous cermet components of engines.

The standard infiltration process used for the production of precision products consists of lightly compacting a briquet of ultrafine (less than 5 μ) titanium carbide powder and sintering at a relatively low temperature to form a strong but very porous compact. This skeleton is an exact replica of the finished product in shape, dimensional tolerance, and surface smoothness. An investment mould of highly refractory material is formed around the skeleton, and subsequently a molten heat-resistant alloy is introduced and contacts the skeleton, usually on top, filling all its interstices by capillary action. Thus, a continuous intertwined two-phase structure is produced upon solidification.

This technique has been modified so that infiltration of the skeleton takes place along preferential paths, resulting in metal-enriched areas at the sensitive edges of the aerofoil and at the root. By careful control of the heating cycle, shrinkage of the skeleton can be directed to create gaps between the interior surfaces of the investment mould and the exterior surfaces of the skeleton. These gaps become filled with the molten heat-resistant alloy during the infiltration process.

PROPERTIES OF HOMOGENEOUS TiC CERMETS

The properties of titanium carbide cermets are influenced to varying degrees by the microstructure of the composite material. Impact resistance is sensitive to the uniformity of the spacing and the degree of separation of

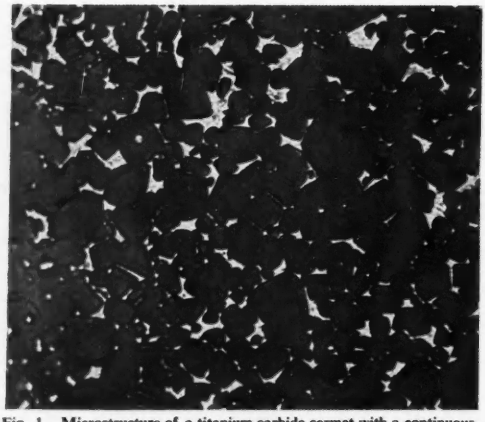


Fig. 1. Microstructure of a titanium carbide cermet with a continuous carbide network. (Unetched, × 1000)

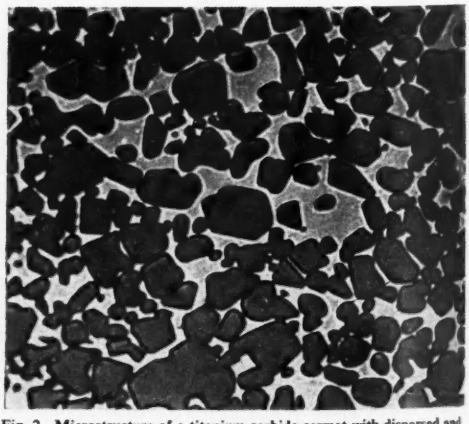


Fig. 2. Microstructure of a titanium carbide cermet with dispersed and isolated carbide grains. (Unetched, × 1000)

the carbide grains. Since impact fractures tend to initiate in and propagate through the brittle carbide phase, a structure of isolated well-rounded carbide grains will have a higher impact strength than a structure with angular coalesced grains. In this connection, Fig. 1 shows a representative structure indicative of low impact strength, where an almost continuous path may be traced through the carbide phase; more than double the impact strength can be achieved by a structure containing more dispersed and isolated carbide grains, such as that shown in Fig. 2.

Unfortunately, stress-to-rupture properties tend to decrease with decreased carbide concentration within certain limits (Fig. 3). The shape of the titanium carbide grains does not seem to affect stress-rupture properties,

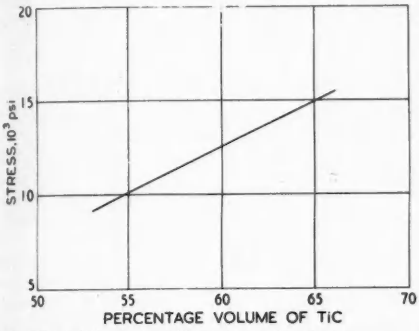


Fig. 3. **Effect of titanium carbide content on the stress to rupture of a cermet in 100 hr at 1800°F.**

but there is evidence that strength decreases with excessive separation. There are also indications that, with the higher carbide concentrations required for high-temperature stress-rupture strength, there is a marked tendency to coalescence of the grains, and this appears incompatible with the degree of carbide-grain isolation necessary for high impact strength. Thus, the achievement of adequate toughness, at various temperature levels up to and above 1800°F, becomes virtually impossible in a homogeneous cermet of the type under consideration.

Graded Components by Infiltration

The problem of combining high impact strength with high stress-to-rupture strength may, however, not be as formidable for an actual turbine component, if the high-impact and high-strength properties are located in those areas where they are most required, i.e., in a suitable graded structure. For example, the aerofoil section of a bucket with a thick dimension in the central area which tapers out to thin leading- and trailing-edge sections could consist of such a graded carbide structure. The heavy load-bearing central sections near the root require primarily high stress-rupture strength, to withstand the centrifugal force caused by the rotating wheel, but need not have an optimum impact strength, since there is enough mass to resist impact fracture; a high carbide concentration may therefore be advantageous. On the other hand, the thin leading and trailing edges should possess the highest possible ductility, because there is only an extremely small amount of mass available to bolster these sections against dynamic stresses and impacts caused by foreign particles. Since the edge sections are not required to carry as heavy a load as the central portion, their structure may incorporate well-separated carbide grains, with a corresponding reduction in stress-rupture strength.

The pores of titanium carbide skeletons can be filled with concentrations ranging from below 50 to above 80% by volume, so that an infiltrated product with a graded carbide concentration can be produced readily. A graded skeleton can, for example, be formed within certain practical limits by pressing a billet with a higher density at the centre, by varying the die fill, or by varying the moulding pressure in selected portions. After pre-sintering, the billet is machined into a turbine-bucket shape whose leading and trailing edges will have a substantially lower carbide density, and this concentration gradient will be maintained after infiltration.

The graded carbide structure can also be achieved to a certain degree on a shaped product, even when the skeleton is of uniform density. With a selected carbide particle size and proper control of the infiltration mechanism, the edges and other high-energy areas of a blade skeleton tend to be infiltrated preferentially with a greater amount of molten metal contacting the surface than the central areas of the skeleton. Owing to this predominance of the liquid metal in the outer areas, dispersion and separation of the carbide grains are preponderant at the edges. This mechanism is also operative, though much less effectively, at the faces of the aerofoil shape.

Nozzle vanes produced with a highly dispersed carbide structure at the edges have withstood a ballistics impact test at the trailing edge without exhibiting the brittle-type failure of conventional cermet materials.

A second important effect achieved by the infiltration process is the integral attachment of a metallic surface film to the carbide body. This metal-enriched surface film is merely an extension of the alloy phase of the infiltrated structure and is saturated with titanium carbide. Since it is produced as an integral part of the carbide body, there is no evidence of a parting line or separation at the transition zone. The structure and bonding of the metal-enriched surface film to the base structure are shown in Fig. 4.

A variation in thickness of the surface film has a marked effect on impact resistance. This was established in a series of impact tests on samples of $\frac{3}{16} \times \frac{3}{16}$ in., which had been cut and ground from infiltrated rectangular bodies, so that one face would have a metal-enriched layer. The specimens so produced were tested with the metal-enriched face of the bar placed in the tension fibre on a pendulum Charpy impact machine, using a 50 in.-lb

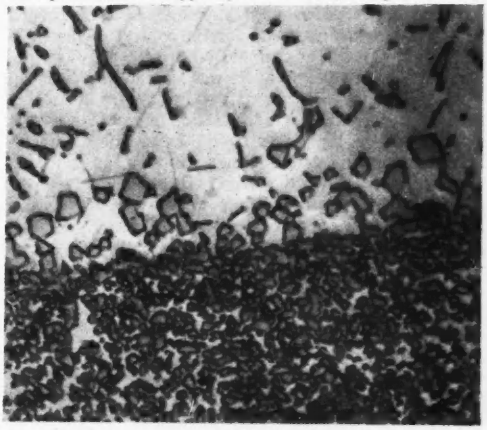


Fig. 4. **Bond between metal-enriched surface film and body of a cermet component. (Unetched, × 200)**

pendulum hammer and a span of 1·05 in. The results are shown in Fig. 5. A surface film of 0·006 to 0·010 in. more than doubled the impact resistance, and a film about 0·025 in. thick increased the impact resistance fourfold to a value of about 30 in.-lb. This improvement may be attributed to the plastic deformation which is achieved at the metal-enriched surface and which greatly reduces the tendency to crack propagation into the core of the test bar.

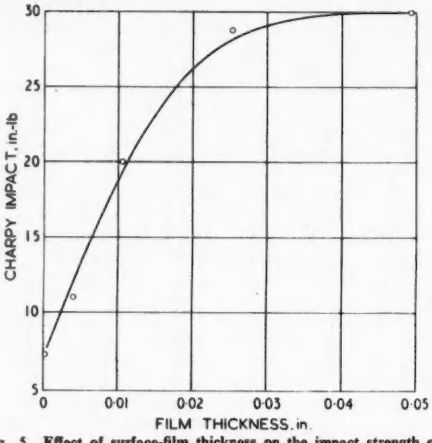


Fig. 5. Effect of surface-film thickness on the impact strength of a titanium carbide cermet.

It is apparent that this method of improving impact resistance may be used to great advantage in high-temperature engine components such as turbine buckets and nozzle vanes. Critical areas subject to ballistic impact, such as the leading and trailing edges of the aerofoil, may be made appreciably more ductile by a metal-enriched layer or encasement.

Metal-Enriched Aerofoil Encasements

Infiltrated turbine buckets have been produced on a pilot-plant production basis with a metal-enriched surface closely controlled to a thickness ranging between approximately 0·01 and 0·03 in. at the leading edges, 0·02 and 0·06 in. at the trailing edges, and 0·002 and 0·01 in. on the concave and convex faces of the aerofoil. Metallic layers of even greater thickness have been achieved experimentally by careful control of the infiltration variables. For example, a porous layer of refractory carbide and alloy powder was applied to a titanium carbide skeleton in preferred areas prior to insertion in the investment mould. On infiltration of the skeleton, the porous surface layer was co-infiltrated and then dissolved by the molten infiltrant, resulting in a metal-enriched surface layer of extensive depth.

Coupled with noticeably improved impact resistance is an equally significant enhancement of the high-temperature oxidation resistance of the metal-enriched casing, as compared with the unprotected carbide body. The depth of penetration of oxide in the metal film after a 24-hr exposure at 1800°F was found to be reduced to about 20%. What appears to be of even greater significance is the observation that the impact resistance remained at the original level as contrasted with unprotected titanium carbide composites, in which it was often found to be nearly destroyed by such treatment. Moreover, it may be expected that, thanks to the better ductility of the metal-enriched film, the high-temperature fatigue and thermal-shock resistance of the infiltrated titanium carbide products would be greatly enhanced.

By forming a continuous frame-like structure around the edges and an encasing envelope around the convex and concave faces of the aerofoil, a serious shortcoming of present titanium carbide-base turbine components might readily be overcome. Engine tests have shown that a castrophic failure of a turbine wheel often will result from the impact fracture of one cermet bucket. Upon impact, the bucket shatters and the flying fragments in turn hit the other cermet buckets on the wheel, causing them to shatter. With the incorporation of a ductile metal envelope in the design of the bucket, incipient cracking on impact of the critical edge areas would be greatly minimized. If it did occur, the fragments would be restrained and held together.

The second serious type of premature cermet bucket failure in engine tests occurs in the roots across the fir-tree serrations or the modified dovetails. Such failures indicate that the stresses in this area are greatly aggravated by the method of attaching the brittle component to the turbine disc. Normally, the ductile root of conventional metallic buckets with fir-tree serrations allows for stress equalization and plastic deformation due to stress raisers caused by misalignment. Here again, the precision infiltration process appears to be the logical answer to the production of titanium carbide buckets having a root which consists either partially or entirely of the ductile, strong super-alloy. By application of the principles outlined for the formation of the metal-enriched aerofoil encasement, it is also possible to form a large metal section at the root by an extension of the infiltrant phase of the structure.

The Future of Infiltrated Components

Graded titanium carbide-base turbine buckets and nozzle vanes should perform satisfactorily in jet engines under normal conditions of operation involving relatively moderate stresses and temperatures. However, there still exists the question of the technical and economic attractiveness of a product of this type, and this must be proved to the satisfaction of jet-engine constructors, before mass production of the component can be contemplated.

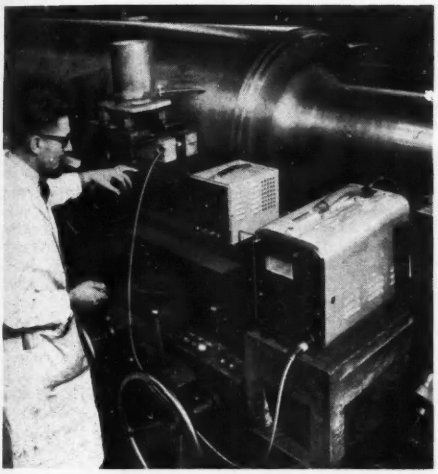
Improvements in the high-temperature strength of certain super-alloys, especially by vacuum melting, make it almost mandatory that cermet-type components should perform up to a temperature noticeably above that of the latest super-alloys developed, i.e., certainly above 1800°F, and preferably up to as high as 2000°F, in the hottest areas of nozzle vanes and bucket aerofoils. Naturally, they would have to perform equally satisfactorily at any intermediate temperature encountered during heating and cooling.

It is the authors' belief that graded components of the type and structure producible by the infiltration technique will answer this requirement, possibly constituting the only practical solution to the problem. If care is taken that the metal-enriched encasement around the aerofoil and root is not impaired by the diffusion of embrittling elements, it should be possible to increase, by suitable addition to the titanium carbide, intrinsic strength and oxidation resistance at elevated temperatures. Not only additions of other carbides, but also borides and silicides, may serve this purpose. As a result, a complex titanium carbide-bearing base structure may be developed which, when infiltrated and encased with the best present super-alloys, will yield jet-engine turbine components useful in the range of extremely high temperatures.

British Industrial Developments

Fault Detection in Rotor Forgings

An automatic method of ultrasonic testing of rotor forgings for internal faults is being investigated by the General Electric Co. Ltd. at their Witton Engineering Works, using a Solus-Schall ultrasonic flaw-detector in conjunction with a monitor. The rotor is mounted in a lathe with the probe of the flaw-detector carried in the tool-rest. As the rotor is slowly revolved, the probe traverses its full length. Any internal fault causes a light to appear on the monitor,the operator marking its position on the forging. Alternatively, provision is made for an audible warning to be given. Subsequently, any faults discovered by this preliminary scanning are explored fully by hand methods, to establish their precise nature.



In addition to the outstanding advantage of a guaranteed 100% coverage, the operator is relieved of the strain imposed by continuously watching a trace on a cathode-ray tube, so that his efficiency is improved. Furthermore, the time necessary to conduct a full examination of the forging is reduced very considerably.

A further development of this system, now being considered, is the provision of electronic apparatus whereby the position of any fault in the rotor is marked automatically, so that it would no longer be necessary for an operator to be in constant attendance. It is expected that this would also enable the speed of scanning to be still further increased.

Improved Turning Rolls

To meet the demand for greater accuracy and better-nuality welding, Donald Ross and Partners Ltd., of Crawley, have introduced new and improved features of their Twinner turning rolls to provide precision, speed of adjustment, and protection of the equipment against overloading and shock-loading. The principal developments provide the refinements necessary for fully automatic welding and include the fitting of a Ross clutch, the mounting of the roller assemblies on flanged wheels, the fitting of hydraulic variable-speed drives, and the availability of speed indicators, etc.

Positive, smooth torque is essential to an efficient turning roll, and it is necessary to protect the transmission against damage, without losing efficiency. Final worm and wormwheel drives on these rolls are now fitted with the new Ross clutch, allowing the rollers to idle when declutched. When loading a vessel, shock is absorbed by the rubber-tyred rollers and cannot be transferred to the gear teeth, eliminating possible damage to the gears. The clutch has no keys and, because engaging members are tapered, provides a positive drive without backlash. There is no danger of losing traction when rotating out-of-balance loads, and the clutch can be engaged in any position, regardless of the relationship between the final gear and roller shaft.

All roller bracket assemblies are now mounted on flanged steel rollers, and the assembly runs on machined ways on the roller bed. Rollers are easily moved into position and secured by locating pins in the holes on the bed. Cranage is no longer required for adjustment, and securing nuts and bolts are unnecessary. In addition to the quick-adjustment roller assemblies, these rolls are stressed to carry 100% overload, as steel overload-discs are assembled on each side of the rubber tyres, and these prevent the tyres being loaded beyond their maximum load capacity.

A hydraulic variable-speed drive is now fitted, providing a wide speed range, ample for welding requirements. The manufacturers can, however, supply equipment with single-speed drives, to suit particular requirements, or equipment to provide a complete electrical remote-control system. Speed indicators, to register the speed of rotation on the rolls, are available for all models, if required.

In addition to these developments, a Model TR.2 and TR.15, of 2 and 15 tons respectively, have been added to the standard range. The complete range now accommodates workpieces from a few hundredweights up to 100 tons, using only one power unit and one idler unit, but equipment for loads up to as much as 200 tons can be supplied.

Heat-Resisting, Non-Melting Industrial Lubricants

Based on " Bentone " greases containing molybdenum disulphide, a new range of greases, with an extremely wide range of industrial applications, is announced by Rocol, Ltd., of Swillington, Leeds. Normally, greases consist of mineral oils thickened with various metallic soaps. In Bentone greases, however, instead of a metallic soap, a modified bentonite clay is used as the thickening agent. These greases are more heat-stable and resistant to working, and have better adhesion to metals, better water-repellent properties, and less oil separation than conventional greases. In addition, they have literally no melting point and can thus be used in conditions up to the point of inflammability of the basic oil. The combination of molybdenum disulphide with these improved greases constitutes the new range of lubricants, which have been designated " Molytone " greases, and which combine the lubricating and pressure-resisting qualities of molybdenum disulphide with the special qualities of the Bentones.

Molytone grease has exceptional temperature-resistant properties, enabling machinery to be operated continuously at temperatures as high as 450°F, or intermittently at still higher temperatures. Its pressure resistance gives positive lubrication at 100,000 psi working pressure, thereby countering high-pressure working conditions normally causing wear, sticking, or seizure. Furthermore, its excellent water-repellent properties permit it to be

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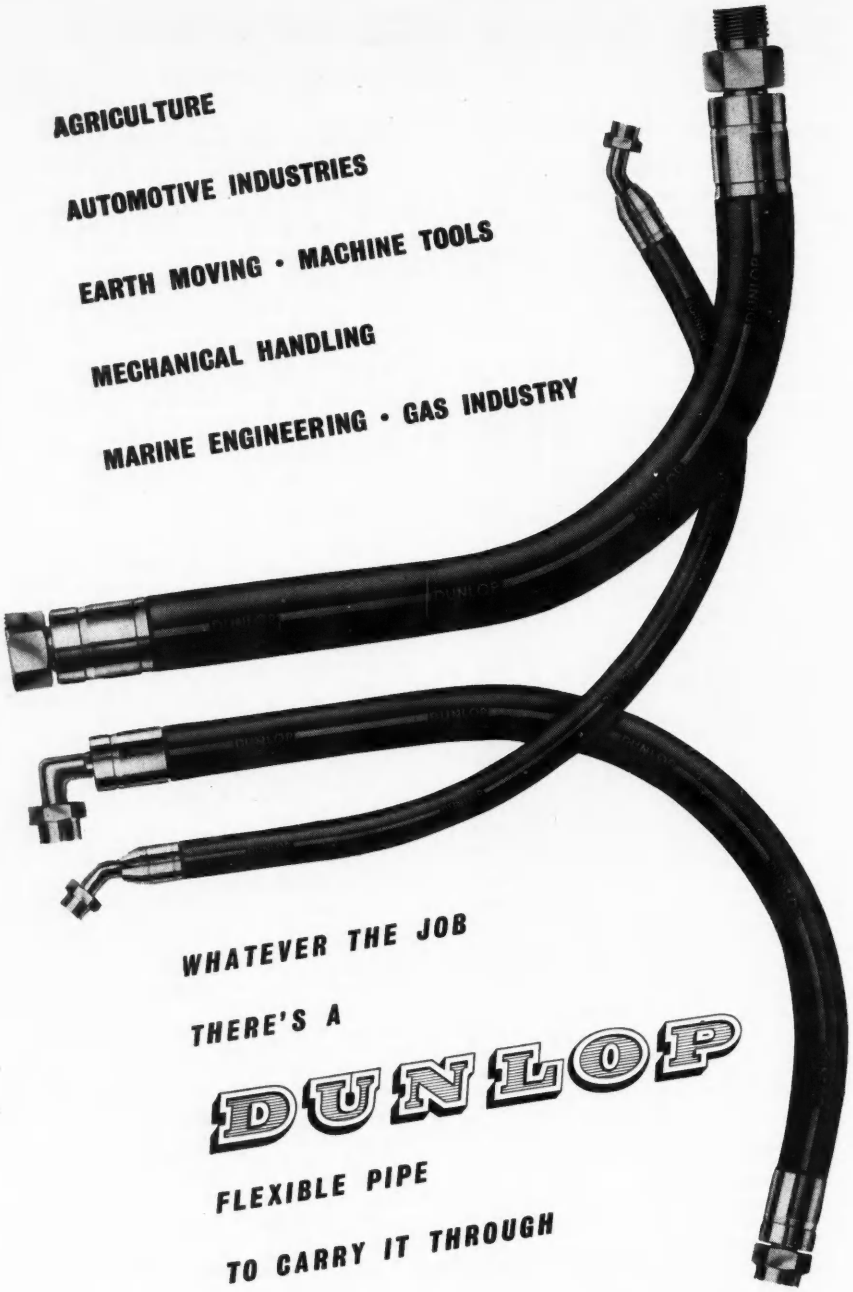

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used in high-pressure steam atmospheres and other wet conditions, without impairing its lubricating properties and with complete absence of " bleeding ". Molytone grease, however, has the same soft consistency as normal ball- and roller-bearing grease.

Molytone L.M. grease, containing a higher content of molybdenum disulphide than Molytone grease, is designed for the lubrication of gas-cocks, plugs, and valves, particularly where considerable heat is experienced. It also has many applications as an anti-seize grease, especially where sealing at high temperatures is required, and for the prevention of freezing and fretting of static parts fitted together or under semi-static conditions.

Molytone X grease, a combination of molybdenum disulphide and a non-petroleum Bentone grease, has a special application as a rubber lubricant. Unlike previous rubber lubricants, it is soft and can be applied by ordinary lubricating equipment. Molytone X does not soften or otherwise harm rubber and, in addition, it has particularly high resistance to petroleum solvents and, unlike other petroleum-resistant greases, it is also water-repellent.

Molytone C grease is a special modification of Molytone X grease ; as such, it is softer and is intended for use in car-brake servomechanisms.

Camshaft Pump with Pneumatic Governor

The Bryce range of " A "-size multi-cylinder camshaft pumps manufactured by Bryce Berger Limited, of Staines, Middlesex, is now available fitted with pneumatic governors. The complete unit, with its light and compact construction, is particularly attractive, in view of the present trend of small diesel-engine design towards smaller and lighter engines running at higher speeds. The " A "-size camshaft pump caters for the higher duties imposed by this trend by employing roller bearings of the same size as those used in the " B "-size range of pumps. This permits the use of a stiff, robust camshaft which is better able to resist dynamic deflections, and which, together with the high load-carrying capacity of the bearings, provides a greatly increased pump life.

The governor unit is capable of controlling the engine speed at any value between maximum and idling speeds, which are predetermined by adjustable stops, and is therefore eminently suitable for commercial-vehicle applications and also for industrial- and marine-engine use. The two main portions of the unit are the venturi air-flow control unit mounted in the induction pipe between the air filter and the engine, and the pressure-sensing unit mounted on the fuel pump and operating the fuel-control rack. These two parts are connected by an airtight tube. The operation of the governor is based on the well-known principle that air velocity across the end of a tube will tend to cause a pressure drop in the tube. In this case, the air velocity to the engine across the top of the tube is controlled by a butterfly valve in the venturi unit, and the pressure drop set up in the tube is transmitted to the governor casing. The volumetric efficiency of the engine is not affected by the presence of the venturi unit.

The governor casing is divided into two halves by a leather diaphragm, specially treated and prepared to render it extremely pliable. One half forms an airtight chamber, with the exception of the tube connection to the venturi, while the other half is open to atmospheric pressure. A light spring acts on the diaphragm, loading the fuel-control rack towards full fuel ; the balance position of the diaphragm, under the action of the spring load and depression on one side and atmospheric pressure on the other, positions the fuel-pump rack so as to deliver the required amount of fuel to the engine. In its operation, the spring also damps out high-frequency pressure variations, thus contributing to a smooth response. In order to stop the engine, a lever is provided which can override the spring and move the fuel-control rack to the "stop" position. A small auxiliary spring is also housed in the governor casing to provide the necessary engine idling performance. Engine speed is controlled within its limits by the lever operating the butterfly valve, which may be actuated by an accelerator pedal or other appropriate mechanism.

Photomicrographic Equipment with Automatic Exposure Mechanism

A new automatic camera microscope, with wide scientific and industrial applications, manufactured by Carl Zeiss, of Western Germany, is announced by the U.K. agents, Degenhardt & Co., of London, W.2. The instrument is suitable for visual observation, as well as for photomicrography, and, in addition, it can be used for microprojection, making it particularly valuable for instructional purposes. For metallurgical work, special wide-angle objectives are available.

The estimation of exposures in photomicrography, especially in colour work, has always been comparatively difficult and has demanded considerable technical skill. An outstanding feature of the new microscope, designated the " Ultraphot II ", is a fully automatic exposure mechanism which completely eliminates this difficulty. A single push-button opens the camera shutter, and the exposure time is then adjusted automatically to the correct value, depending on the intensity of illumination, after which the shutter closes by itself. The camera can be used for photographs up to 9 × 12 cm on plates and flat film, and its length can be varied by as much as 30 cm by simple rotation of a knob. Comprehensive survey photographs with photomicrographic objectives can be produced with specimens up to 60 mm in diameter. The scale of the focused image can readily and speedily be ascertained by means of a gauge number. All operating control-knobs are readily accessible, and are directly in view of the operator. The controls for coarse and fine adjustment are located within easy reach, while the push-buttons for the automatic camera mechanism are

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mounted on the front of the base-plate. The nstrument is supplied complete with a specially constructed, sturdy bench, designed to provide a solid vibration-free support and to enable the operator to assume the most favourable and comfortable posture.

Bright-field and dark-field methods and phase-contrast techniques, with either trans- or epi-illumination, employing polarized or non-polarized light, may readily be used. Change-over from one technique to another is extremely simple, mounting of the various accessories being easily and rapidly effected. In addition, the intensity of illumination can be varied and adjusted to meet even the most difficult problems, e.g., fluorescent light.

Pivoted Storage Units

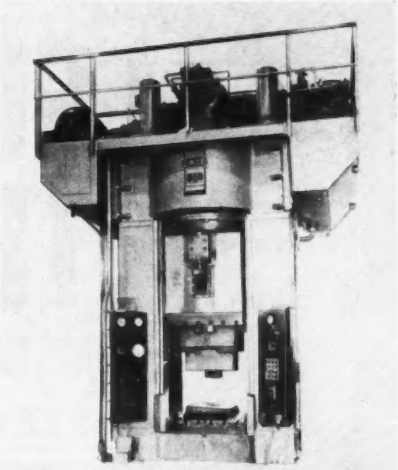
Stated to provide the utmost possible economical storage capacity per square foot of floor space and to revolutionize present storage methods, the "Pivoted Wing" storage units manufactured by Carbidall Ltd., of Blackburn, Lancs., are designed to cut component, accessory, or other storage space by up to 75%. These units also achieve maximum accessibility, together with maximum economy and free movement of trucks and pallets for fast loading and unloading. Furthermore, they are equally efficient for fast- or slow-moving stocks.

The system comprises a number of unit storage sections (fitted with bins or shelving) mounted on three wheels and pivoted within a container framework. A gentle pull causes the unit to glide out from the container in an arc, while a push serves to swing it into the closed position. Units will hold loads from a few pounds to one ton, and can be supplied air-conditioned, fireproof, dust-proof, and theftproof. In addition, they can be adapted to marine purposes and to vehicle mounting for mobile shops, libraries, etc.

Hydraulic Forging Press with Electromagnetic Controls

A new 1500-ton hydraulic forging press, manufactured by the Erie Foundry Company of America, is announced by Burton, Griffiths & Co., Limited, of Kitts Green, Birmingham, sole agents in the U.K.

Features of this press, which is designed for no-draft, non-ferrous forgings, include the provision of electromagnetic controls which permit instant response to delicate fingertip control by the operator. All controls are grouped on one accessible control panel, and the



press is arranged for semi-automatic operation, so that the forging cycle can be preset to incorporate any desired advance, pressing, release, and return speed and tonnage. The press can also be operated manually, and this manual control is readily detached from the control panel by the operator, who is then free to control the press from any nearby position. This remote feature is particularly valuable when checking die alignments from all four sides of the press. Hydraulic knock-outs are used, and these can be set to eject forgings from the die at any position in the cycle. The guide system is easily adjustable to ensure accurate die alignment. No shimming is required, and adjustment can be made in one hour only.

The press, which is deflection-free, rigidly built, and self-contained, has an opening of 60 in., a main ram diameter of 38 in., and a stroke of 48 in., the speed of advance being 400 ipm, the pressing speed 0 to 27 ipm, the release speed 0 to 27 ipm, and the return speed 360 ipm. Other sizes, with modifications for any hydraulic-press forging requirement, are also available.

Electrodynamic Standards for Vibration Pick-ups

(*Concluded from page 72*)

standardizing the calibrator and calibrating vibration pick-ups, care is taken to avoid frequencies at which transverse motion is present.

Calibrator Performance

Several electrodynamic calibrators standardized by the reciprocity method are now in service at the Bureau for the calibration of vibration pick-ups. A redetermination of the constants a and b for one of the NBS calibrators has shown no significant change after the lapse of a year. With respect to frequency, the calibration factors of the standards have been found to be constant to within 1% up to 900 cps. At higher frequencies the calibration factor is known with somewhat less accuracy, because of flexibility in the standard as a result of longitudinal resonances. This flexibility changes the electromechanical characteristics of the standards and results in less stability at the higher frequencies. Plans are under way to modify the standards to achieve an accuracy of 1% at higher frequencies.

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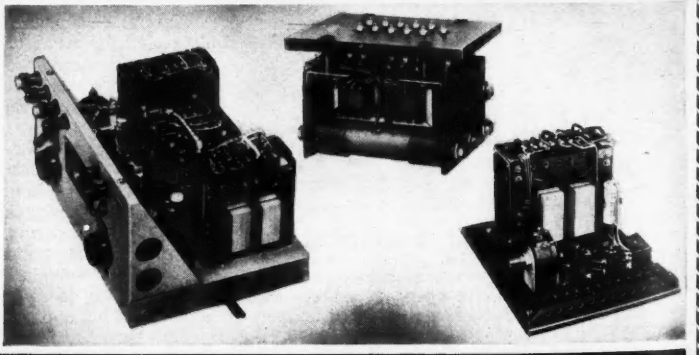

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ADDITIONAL ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

Future Developments of the High-Speed Diesel Engine.
By E. CHATTERTON. (From a paper presented at the annual meeting of the *American Society of Mechanical Engineers*, November 28, 1956, 20 pages, 17 illustrations.)

It is well known that an engine with multiple cylinders of small dimensions running at a high speed can be made lighter and smaller for a given power output than an engine having a small number of cylinders running at lower speed. For the purposes of this paper, a high-speed diesel engine is one which is designed specifically with the object of exploiting this fact and achieving the minimum weight and space occupied per horsepower produced. In this connection, there are many applications for diesel engines where minimum size and weight are important, e.g., in the propulsion of certain naval and similar craft, and in portable or mobile equipment of various kinds. In such applications, high-speed engines are essential, if technical requirements are to be fulfilled. There are, however, other applications for which heavier and larger types of engine are traditionally employed, but which would show considerable technical and economic benefit if smaller and lighter engines were used, and two typical examples are considered in this paper, which has three main objectives, i.e., (1) to show that the development of the high-speed diesel engine is justified on both technical and economic grounds; (2) to explore the lines of development which at present appear to offer the greatest possibilities of reduction in engine size and weight, and to consider existing limitations; and (3) to assess the relative suitability of various basic types of engine for such developments.

This paper represents what may well be the first attempt to predict from first principles the practical performance possibilities arising from the combination of a highly supercharged diesel engine with a gas turbine, when operating at high values of air/fuel ratio, with particular reference to "compounding", as exemplified in the Napier "Nomad" and now in development as applied to the "Deltic". After stating briefly a case for the high-speed engine, the paper formulates a basis for performance assessments, which uses fairly standard methods but which is essential to support the conclusions derived. In addition, the actual methods of performance computation for the various cylinder arrangements are explained, and a basis stated for the prediction of resulting piston temperatures, which has been found to agree with experimental recordings. Finally, limitations are discussed and conclusions drawn regarding the type of engine best suited to such developments. In this connection, some of the deductions made from performance computations are as follows:—

(1) Despite its economy in the use of air, the four-cycle engine cannot compete on the basis of brake horsepower per cubic inch of cylinder volume with the two-cycle engine.

(2) Because of its small piston diameter in relation to its cylinder volume, the opposed-piston cylinder is able to produce higher powers for a given cylinder diameter than the Uniflow engine for the same piston temperatures.

(3) The efficiency of the supercharger exerts a very considerable influence on the final power output, particularly when high degrees of supercharging are used. For high manifold pressures, therefore, an axial-flow supercharger may be regarded as essential.

(4) The turbine is capable of producing more power than that absorbed by the supercharger. Consequently, to take full advantage of this excess power, the turbine must be geared back into the engine output, thereby forming a "compound" engine. In the free-running turbocharger, the turbine power is limited to that demanded by the supercharger, involving sacrifices in total power output.

(5) Using a piston of 5 in. diameter, the brake horsepower per cubic inch of volume can, by supercharging, be increased to three times the value at zero manifold pressure, with no increase in piston temperature.

(6) The increased power resulting from increased supercharging and air/fuel ratio can be achieved with very little sacrifice in specific fuel consumption.

It is finally concluded that the greatest possibility for reduction in engine size and weight is offered by a highly supercharged two-cycle engine compounded with a turbine driving an axial-flow compressor, and that the cylinder layout which offers the greatest potentiality for such development is the opposed-piston design.

The Effect of Temperature on the Fracturing Behaviour of Mild Steel.
By J. D. LUBAHN. (From *The Welding Journal, U.S.A.*, Vol. 35, No. 11, November 1956, pp. 557s–568s, 18 illustrations.)

THE primary objective of the investigation described in this paper was to observe ductility and mode of crack propagation as a function of temperature for several kinds of tests which subject the metal to different combinations of a number of variables, including temperature, strain rate, and stress state (relative magnitude of the three principal stresses). A secondary objective was to obtain a better understanding of the nature and significance of the notched-bar impact test.

Samples of 4-in. mild-steel plate were tested in various ways, and various fracture properties were observed, i.e., strength, energy-absorbing capacity, ductility (local plastic strain preceding crack initiation), and amount of gradual tearing preceding brittle crack propagation by cleavage. Each property was measured for various temperatures, loading rates, and biaxialities for both notched and unnotched specimens cut in both the rolling and thickness directions of the plate.

At temperatures above + 150°C the ductility, strength, and energy-absorbing capacity are large and the crack propagates entirely by gradual tearing (fibrous). As the temperature is lowered, sudden crack propagation by cleavage appears and the energy-absorbing capacity begins to fall. At + 30 to — 40°C, depending on the loading rate, fibrous fracture has disappeared and the energy values are quite low, but the strength is still good and the ductility is still 50%, even in notched specimens. The ductility becomes very small at — 100 to — 180°C, depending on the strain rate, biaxiality, and presence of a notch (triaxiality). The slow notch bend strength, based on crack initiation, rises continuously with decreasing temperature to — 150°C and then drops off slightly. At temperatures below — 80°C, complete separation occurs when the first crack appears. At all higher temperatures, the bending load rises while the crack gradually tears open, resulting in a fictitious "strength" value, if strength is based on maximum load. Maximum load is 50% higher than the load for crack initiation for temperatures above about — 20°C.

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The notch bend ductility is the same as the biaxial unnotched bend ductility in the neighbourhood of room temperature, but much less at — 60 to — 120°C. This result is discussed in terms of triaxiality (tension in all three principal directions) and the location of crack initiation. The ductility in the normal direction is much lower and more erratic than in the rolling direction, but the strengths in the two directions are essentially the same, except below — 150°C.

The results show that transition temperature in a Charpy impact test is related to the temperature range in which the metal is losing its ability to tear gradually (without cleavage), while transition temperature in an unnotched impact bend test is related to the temperature range in which the metal is losing its ductility. The latter temperature is about 100°C lower than the former temperature.

There is nothing about these particular test results to indicate that mild steel would be either weak or brittle at temperatures above — 50°C, i.e., under normal conditions of service. Catastrophic service failures are discussed in terms of other factors not present in these tests, such as size, notch sharpness, and secondary loads.

An Experimental Treatment of Non-Linear Servomechanisms.

By R. S. NEISWANDER. (From *Applications and Industry* (*A.I.E.E.*), *U.S.A.*. No. 27, November 1956, pp. 308–316, 10 illustrations.)

IN practice, servomechanisms are intrinsically complicated. Amplifiers saturate, electric and hydraulic actuators are not linear, and the efficient despatching of an instability by an ingenious network simply creates several new problems when the new stability limit is pushed. Graphic or analytic methods become tedious, and eventually experimental methods have to be used. This transition to an experimental technique has one fundamentally important consequence, i.e., the system itself need no longer be approximate. An approximation of the actual system input, somewhat resembling that used by more elementary treatments, is retained, to allow the formation of rules of system performance and of effecting improvements. The experimental technique cannot be couched in rigorous mathematics for, indeed, it is the weakness of the mathematics that has demanded the experimental tests.

In this paper, an experimental method of achieving appropriate first switching points for certain non-linear single-loop servomechanisms is described. The servomechanism is assumed to be a positioning device. For control intelligence, non-linear corrective functions of output velocity, static error, or error velocity are assumed continuously available. Since the method is experimental, either the actual device or a reasonably accurate analogue model can be used, and thus is not subjected to the same limitations as those imposed by simplified mathematical models. Here, the basic device may be of high order and include many unintentional non-linearities. One principal limitation accepted by the method is that the system input can be described by a limited Taylor series, i.e., at time $t = 0$; the quiescent system is confronted with an input consisting of an arbitrary position step plus an arbitrary velocity step, or ramp, plus an arbitrary acceleration step, etc. Switching points subsequent to the first are assumed to be compacted into the terminal region of the response, either to be absorbed by allowing the terminal region to be unsaturated, and perhaps linear, or to be determined by other methods. This assumption might be interpreted in practice as the requirement that the system components such as amplifiers, actuators, and sensing elements be in themselves relatively well damped. A highly underdamped element that tends to " wiggle " or ring during the positioning operation might well require a more elaborate experimental treatment.

The method described in this paper is concerned with attaining physically limited (or saturated) responses. It is a simple, two-step procedure, in which a set of system responses is recorded, and from these a " corrective " function is directly measured, establishing the first reversal point. Near the null region, additional reversals are absorbed by permitting the unsaturated null region to be reasonably linear. The method relies on a work sheet which has the property of directly relating the transient responses of the system with intentionally added corrective functions. The work sheet is a system-response plot of output velocity against system static error, the mathematical implications of this plot as a phase-space cross-section being ignored.

The Distribution of Surface Pressure over the Friction Lining of Brakes with Rigid Shoes and Drums, and their Efficiency.

By H. SCHRODER. (From *Acta Polytechnica, Sweden*, Mechanical Engineering Series, Vol. 3, No. 8, 1956, 44 pages, 11 illustrations.)

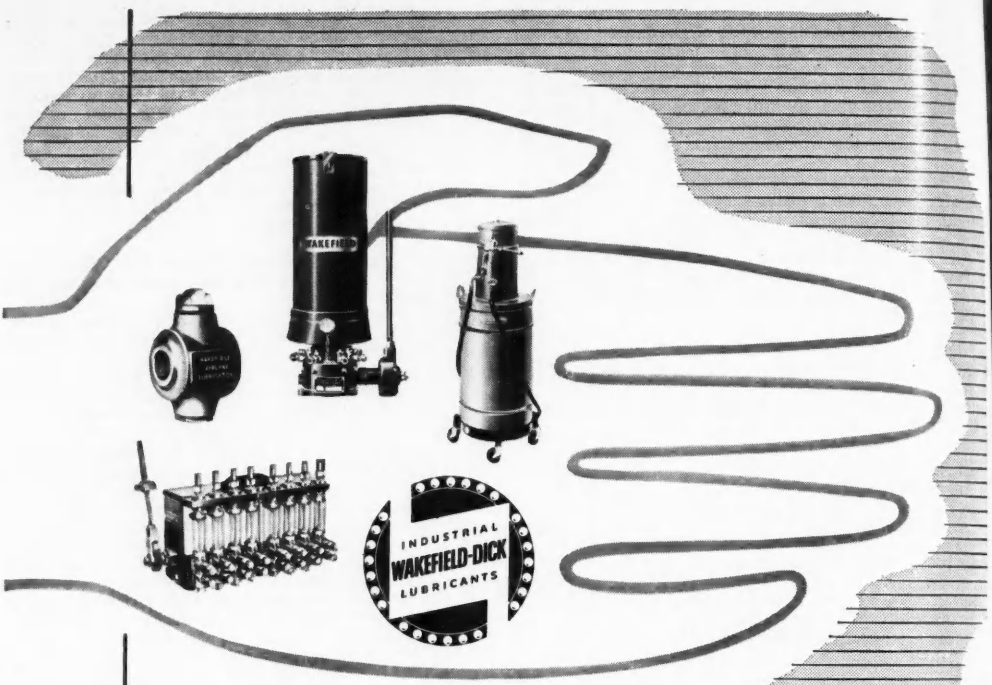
FROM the technical literature on the subject, it appears that the coefficient of friction of a brake lining frequently varies with the magnitude of the surface pressure exerted when the lining is forced against a rigid brake drum and with the temperature rise produced in the brake drum by a braking moment. It also appears that the higher is the surface pressure, the heavier is the wear on the lining. It was therefore considered appropriate to attempt to form an idea of the distribution of the surface pressure over a brake shoe with a lining of even thickness, and of this distribution when the lining has been worn in such a way that its thickness is smallest at the centre of the shoe and greatest at the ends of the lining. In addition, it was considered desirable to investigate whether possibilities exist of distributing the surface pressure more evenly over the circumference of the brake shoe than has been possible with conventional brake shoes. In connection with such investigations, it would be natural to calculate the change in braking moment with increasing brake-lining wear. Finally, it seemed reasonable to try to develop formulae for comparing the braking moments of the braking systems most generally used, assuming the same dimensions of brake linings, the same coefficient of friction, and the same operating force.

In this paper, it is shown how equations found for the determination of surface-pressure distribution over the brake drum are used for determining the ratio between maximum surface pressures, as well as between braking moments for worn and unworn linings. Furthermore, the same equations are applied to the determination of the ratio between the efficiencies of different types of brakes when activated by a force of the same magnitude. As the efficiency of a brake depends on whether the heat developed by the brake can be conducted away, this problem has also been investigated.

The Mechanism of Oil Loss Past Pistons.

By P. DE K. DYKES. (From a paper presented at a meeting of *The Institution of Mechanical Engineers, London*, January 25, 1957, 16 pages, 16 illustrations.)

INSOFAR as the mechanism of oil loss past pistons is concerned, it would appear that, in the past, there has been no generally accepted and reasonably complete explanation of this mechanism. Since this mechanism has been lacking, the attainment of a good oil consumption has been a matter of trial and error, based on previous designs which have been found to be satisfactory in operation. Such a process is often inefficient, unless it is supplemented by a knowledge of the mechanisms involved. Some work had been done, using an engine which was motored without combustion, its cylinder being of glass to permit visual observation of oil, piston, and rings by stroboscopic and photographic means, and it was concluded that a major portion of the oil consumption occurs when the top ring lifts axially in its groove. However, other work has demonstrated the



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extreme sensitivity of the oil consumption to the angle between the periphery of a ring and the cylinder wall, and suggests that the major route for oil leakage is past the ring periphery.

In view of the lack of a satisfactory explanation of the mechanism of oil loss past pistons, it was decided to undertake the further research described in this paper, and also to use a laboratory test rig, instead of an actual engine. One reason for using a test rig was that it could be designed so that any unwanted variables could be eliminated and the remaining variables brought individually under the control of an operator, thus making the results far easier to analyse than those from an engine. Another reason was that it might be possible in a test rig to examine individually each of the suspected routes for the loss, instead of their total effect.

As a result of this investigation in the laboratory test rig, in which a piston was both reciprocated and subjected to gas pressures, the following facts have been brought to light, and it is believed that this is the first time that they have been demonstrated:—

(1) Oil control is a two-way and not a one-way problem. The only reason why the rate of oil consumption is reasonably small in most engines is that, although a large amount of oil flows towards the combustion chamber past the rings during part of the engine cycle, nearly all of it returns to the sump during the remainder of the cycle.

(2) The various routes permit surprisingly large rates of oil flow in both directions past rings. This applies even with good rings.

(3) If oil is added to the combustion chamber, a good ring arrangement will pass it to the sump at a rate up to many thousand times larger than the normal rate of oil consumption with the same arrangement.

(4) The piston lands, and the top land in particular, play an important role in deciding the rate of oil consumption. If some oil has passed the rings, it may be thrown into the combustion chamber and be lost; alternatively, an effective top land may retain this oil until it is returned to the sump.

Transistorized Indicator for Measuring Jet-Exhaust Temperature.

By G. H. Cole. (From *Electronics*, *U.S.A.*, Vol 29, No. 12, December 1956, pp. 143–145, 3 illustrations.)

Accurate measurement of jet-engine exhaust-gas temperature permits protection against excessive engine temperatures without sacrificing performance as a result of conservative operation. Thermocouples are capable of withstanding high temperatures and corrosive atmospheres and permit averaging of temperature around the exhaust area. However, their use in precision systems produces problems in amplifying low-level d.c. voltage. The most feasible approach has been to compare the thermocouple signal with a d.c. reference voltage and to convert the difference signal to a.c. for amplification. When this technique has been used in conjunction with a null-rebalance servo-loop, accuracies of about $\pm$ 0·5% for temperature ranges of 1000°C have been achieved.

A typical electron-tube or magnetic-amplifier system, of reduced size and sufficiently robust construction for airborne use, consists of a panel indicator, 2 in. in diameter and 4 in. in length, and a remotely located amplifier occupying about $4 \times 4 \times 4$ in. The volume of this two-unit system is thus about 75 cu in., and its total weight is of the order of 5 lb. The equipment described in this paper uses semiconductor devices for reference-voltage supply, a thermocouple bridge, a modulator, and an amplifier. The entire circuit is mounted in a 2-in. diameter case, $5\frac{1}{2}$ in. in length, weight and space requirements being about one-fifth those of vacuum-tube or magnetic-amplifier equipment. The components, which are assembled on five circular, stacked cards, are pre-assembled on the five cards, which are then mechanically fastened and wired together in the final amplifier assembly. In laying out the circuit and mounting components, it was found essential to mount certain components not only in exact locations, but also with a specific orientation, to reduce pick-up.

Because of the use of miniature components throughout, the entire assembly is rugged. The case is hermetically sealed after calibration adjustments are made, as the low ageing properties of the components involved do not require re-adjustment. The accuracy of the system has proved to be as good as that of larger vacuum-tube units. Under room-temperature conditions, the error is less than $\pm$5°C across the 1000°C range ; under environmental extremes, less than $\pm$ 12°C error occurs. The travel time of the pointer across the 1000°C range is less than 3 sec, and response to a step input occurs with negligible overshoot. The power consumption of the system is 7 W. One result of transistorization is the almost complete lack of sensitivity of the high-gain circuit to pick-up from external sources.

A Strain-Gauge Method of Measuring Strains in Adhesive Bonds.

By C. B. Norris, W. L. James, and J. T. Drow. (From *ASTM Bulletin*, *U.S.A.*, No. 218, December 1956, pp. 40–49, 15 illustrations.)

The strength exhibited by adhesives as they exist in bonds may be much greater than that exhibited by cast specimens of the same adhesives. This difference may have two causes, both related to the thickness of the adhesive film composing the bond. Firstly, the deformations in the adhesive are restrained by the proximity of the adherents, and, secondly, a relatively large percentage of the adhesive has special physical properties, because it has become solid in the neighbourhood of an interface. Thus, in the study of the mechanism of adhesion described in this paper, measurements were made of the strain in the direction of the thickness of a bond due to a tensile stress in the same direction.

The thickness of an adhesive bond may be about 0·001 in., and the strain at failure may not be more than 2%. To obtain smooth, reproducible stress-strain curves, the strain should be measured in units not greater than 0·001%; in other words, a change of 10^{-8} in. (2·5 Å) in the thickness of the adhesive bond should be measured with reasonable accuracy. Mechanical-optical strain gauges at present available are not sufficiently sensitive for this purpose. Their smallest reading is about 10^{-6} in., and it does not seem likely that refinements in their design will greatly increase their sensitivity. Furthermore, such gauges would have to be mounted on the bonded pieces at two points—a large distance from the bond, compared with its thickness—and would at best measure only the relative displacement of one of these points to the other. This displacement probably would not be equal to the average displacement of similar points within the specimen, because of the perturbation of the strain pattern at the edge of the bond. Reasonable measurements might, however, be obtained by this method if the strain gauge were mounted to bridge several bonded joints, and corrections made for the deformations of the several pieces of adherend.

The method described in this paper uses the specimen as a capacitor in a resonant electric circuit and measures changes in the capacitance by measuring changes in the resonant frequency of the circuit. Because of the small modulus of elasticity of the adhesive, compared with that of the adherend, the strain in the adhesive is substantially unidirectional. Thus, the change in the thickness of the bond is substantially proportional to the change in the volume of the adhesive. Accurate evaluations of strain can be made, if the effect of changes in glue-line thickness on the dielectric constant can be determined. For some adhesives, this effect may be calculated, using the Lorenz-Lorentz equation relating dielectric constant to density, when density is assumed to vary inversely with the thickness of the glue bond.

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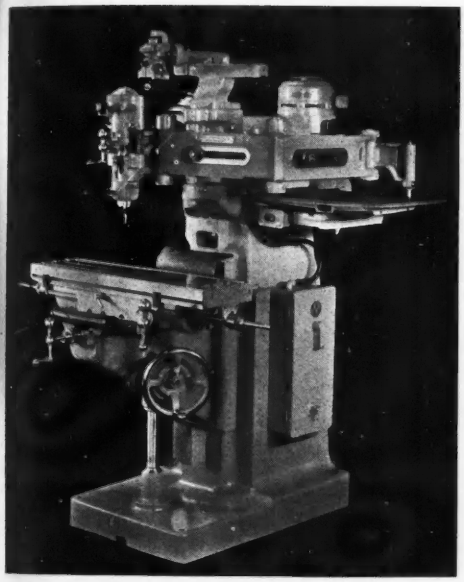
New Materials, Processes, and Equipment

TRACER-CONTROLLED TWO-DIMENSIONAL PANTOGRAPH ENGRAVING MACHINE

A new tracer-controlled, two-dimensional pantograph engraving machine for general engraving, die and mould work, steel stamp cutting or profiling on flat or curved surfaces, and accommodating forming guides and roll attachment, has been developed by the George Gorton Machine Company, of Racine, Wisconsin. The cutter head of the new machine is extremely rigid and is integral with one of the pantograph bars, thereby eliminating the four bearings and two joints normally present in lighter two-dimensional machines. All bars are of heavy-section nickel semi-steel castings, strongly ribbed.



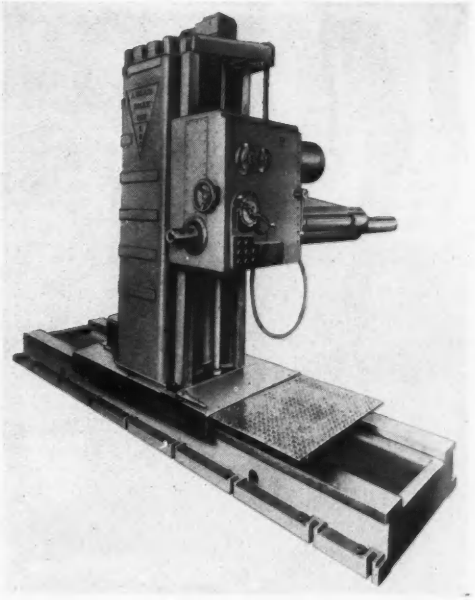
Ratios from 1:1 down to 1 : 6 are obtainable, the pantograph bars being graduated with reductions of 1, 2, 3, 4, 5, and 6 only. Intermediate reductions are easily obtained by using a built-in scale, with increments of 0·02 in., and a formula supplied with the instruction book. Areas are covered by the cutter point at reductions from 1:1 to 1 : 6. The design of the machine is such that it does not restrict the area covered by the cutter when using a forming guide. The heavy-duty spindle is machined from solid bars of a special chrome-manganese alloy steel, hardened and ground inside and out ; it is mounted on preloaded super-precision ball bearings and features a depth-stop graduated in thousandths. Spindle feed is 1 in., plus a 1-in. floating action when using a forming guide. The spindle is ¾ in. in diameter and is of the panto-collet type for straight-shank cutters.

Because of its rugged construction and the location of the master table, the machine can easily be modified for special work. For short runs, simple masters and manual operation provide excellent results. When production runs are required, the machine can often be fitted with a motorized master, automatic down-feed for the cutter, and automatic indexing or positioning of the workpiece, thereby making such applications completely automatic within the cutting cycle. Important features are a quick change-over from motorized masters to manually traced masters and *vice versa*, and the provision of an adjustable ram which can be brought forward over the table up to 10 in. This is especially valuable when milling multiple cavities in large dies and moulds from a single-cavity master.

UNIVERSAL HORIZONTAL DRILLING, TAPPING, AND BORING MACHINE

A new universal horizontal drilling, tapping, and boring machine, incorporating automatic speed and feed selection controlled from a single rotary-action, direct-reading dial, has recently been developed by The Kaukauna Machine Corporation, of Kaukauna, Wisconsin. Of unusual power and rigidity, and providing high accuracy and a greater range of operations, the machine is fitted with a spindle drive motor of from 10 to 25 hp, equipped with a dynamic brake, all movements being activated from a centralized push-button control panel. A 4- or 5-in. diameter nitralloy spindle is available, providing 42 in. of continuous spindle travel. The design is simple and provides efficient high-speed operation on large or small workpieces and on short- or long-run jobs, while extreme column rigidity and power positioning permit the driving of boring fixtures or efficient drilling, reaming, tapping, and spot-facing at the highest feeds and speeds possible with modern cutting tools.

All controls are grouped at the front of the head within easy reach of the operator. The head is completely counterweighted, and an electrically interlocked cam-action clamp locks the headstock to the column. The speed and feed change-gears are of heat-treated, flame-hardened alloy steel, mounted on horizontal heat-treated splined shafts turning in ball bearings. A direct-reading automatic shifting mechanism provides instant gear change, 18 speed changes and 9 feed changes being available, in each case actuated by a single dial.



Observe!

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More important, you can get at the equipment from both sides — not just the back. An unusual combination of extreme accessibility with great economy in space. In other ways, too, an Austinlite control cubicle is an ingenious box of tricks — and may control anything from a complex automatic machine to an all-electric lighthouse. Much of the equipment of these cubicles is itself built by Austinlite — designed in the first place to do a job for which no existing equipment quite fitted the bill. You, perhaps, have a special query that might start a new ball rolling?

Austinlite **EQUIPMENT**

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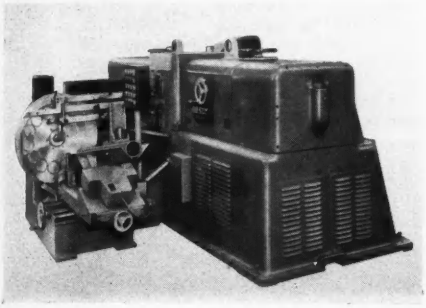
THE MAKERS OF SUMO PUMPS AND STONE-CHANCE LIGHTHOUSES. 28 ST. JAMES'S SQUARE LONDON, S.W.1. TEL: TRAFALGAR 1954

Horizontal movement of the column along the runway is power-operated by a gear motor and worm reduction coupled to a two-speed gear drive, with a traverse of 100 ipm and positioning at 4 ipm. Vertical movement of the headstock is effected by a torque motor coupled to a two-speed gear drive and elevating screw, with the same traversing and positioning speeds as the horizontal movement. This two-speed power positioning operates from a single push-button, ensuring rapid and accurate tool setting.

GRINDING MACHINE WITH PUSH-BUTTON CONTROLS

A new grinding machine, the Besly No. 240-30″ double horizontal disc grinder, with all spindle motors contained within a low, massive, and easily accessible base, and with push-button controls on both sides, has been developed by The Besly-Welles Corporation, of South Beloit, Indiana. By virtue of its improved design, it is stated that maintenance, dressing, changes, and set-up times are reduced by as much as one-third.

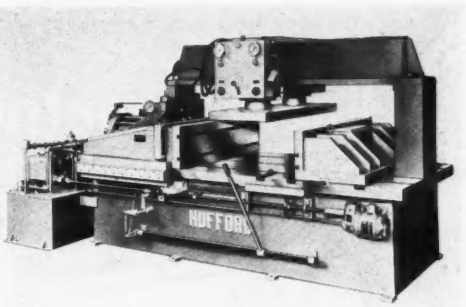


Features of the machine include, in addition to push-button controls and easy reading of head and disc alignment, a sealed spindle quill construction, automatic push-button dressing, automatic sizing to adjust for disc wear, edge-grain Formica wearing ways, fast magnetic through-feed, easy disc removal, and a head assembly sealed with neoprene against dust and coolant. The machine is stated to be very accurate (0·0001 in. for parallelism) and extremely suitable for high-speed precision finishing of piston rings, bearing rings, automatic transmission plates, and similar shapes.

HIGH-SPEED HYDRAULIC DIE PRESS

A new 150-ton hydraulic die press, with vertical die pressure-plate clamping cylinders, specially designed for high-speed forming on dies employing pressure plates, has been developed by Hufford Machine Works, Inc., of El Segundo, California. In this machine, two special 25-ton vertical cylinders supply top pressure, securing the pressure plates in position during forming and overcoming all possibility of deflection. Pressure may be instantly released to permit quick, easy repositioning or complete removal of the workpiece from the die, without the necessity for time-consuming hand clamping. The entire vertical clamping assembly is mounted on rollers, permitting longitudinal location of the clamping unit. Each clamping cylinder is independently controlled from the panel face by a four-way selector valve, pressure being varied, as required, by means of reducing valves. If desired, the overhead beam and clamping unit may be quickly removed simply by unbolting from the frame, increasing the vertical capacity. Eye-bolts are installed to facilitate removal.

A threaded lead-screw type of shipper rod extends along the side of the machine, connecting directly to a variable-volume pump powering the pressing cylinder. The output of this pump is varied from zero to maximum by shifting the rod longitudinally, both manual and power



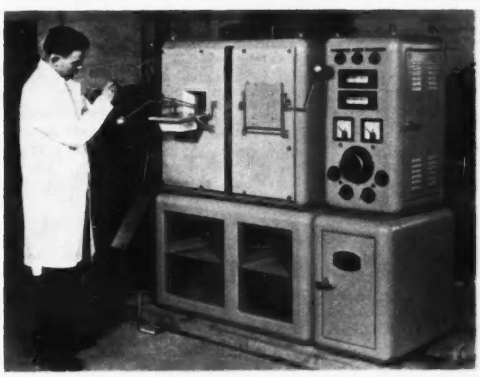
control being provided. Rapid traverse is accomplished by a reversing-type motor, which rotates the threaded shipper rod, causing a stop, threaded to this screw, to be positioned rapidly along the rod. As the ram advances and engages the stop, the shipper rod is thus shifted, gradually bringing the ram to a halt.

Manual control with micrometer adjustment is provided via a graduated handwheel coupled to the screw. Each rotation of the handwheel represents 0·125 in. of ram travel. Both handwheel and starting lever slide on their respective shafts for most convenient positioning. The complete mechanism ensures easy control over the full extent of ram travel, with extremely high repetitive accuracy. A ram-position indicator shows the position of the ram to 0·010 in. at all times, measuring from the starting position.

DOUBLE-CHAMBER HIGH-SPEED STEEL HEAT-TREATING FURNACE

The new Royce electric double-chamber high-speed furnace announced by Royce Electric Furnaces Ltd., of Walton-on-Thames, Surrey, has been specially designed for the precision heat treatment of high-speed steel tools and similar applications. The furnace has two chambers, one operating at temperatures up to 1000°C for pre-heating and tempering, and the other at temperatures up to 1350°C for hardening. Each chamber is 4½ in. wide × 4 in. high × 11 in. long, and is arranged side by side in one furnace body, providing the easy and rapid manipulation from one chamber to the other essential in heat-treatment work of this kind.

A protective atmosphere for the high-temperature chamber is provided by a carborundum muffle, producing, when heated, a neutral atmosphere which prevents scale. This muffle is interchangeable and has a working life of from 50 to 100 hr. No gas plant or drip-feed device is necessary. The high-temperature chamber has long-life silicon carbide resistance heating elements arranged



Facts from the record

of Goodyear conveyor belting

STACKER

APPLICATIONS

A super quality belting for use where there is severe and frequent abrasion and flexing. It is ideal for conveying ores, stone, coke, glass cullet and other highly abrasive materials.

PHYSICAL PROPERTIES

Stacker exceeds British Standards Grade "A" in all requirements of fabric and rubber and may be used at temperatures up to 200 deg. F.
Made with the highest grade friction stock between plies for maximum flex-life.
Has the toughest cover obtainable, with the highest possible resistance to cutting, gouging and abrasion.
Completely inhibited against mildew damage.

AVAILABLE

Stacker is available in 28, 32, 36, 42 and 48 oz. duck weights, from 3 to 9 plies and in a full range of widths, lengths and cover gauges.

STYLE ORS

APPLICATIONS

For operations where extreme resistance to oil is necessary. Especially suitable for carrying oil-sprayed co
meat, fish or food where an

PHYSICAL PROP

STYLE HRS

APPLICATIONS

Specially made for conveying hot, highly abrasive materials when the combination of resistance to high temperatures, extreme abrasion and severe flexing are essential.

PHYSICAL PROPERTIES

The carcass, cover and skim-coating between plies give all the heavy-duty characteristics of a Grade "A"

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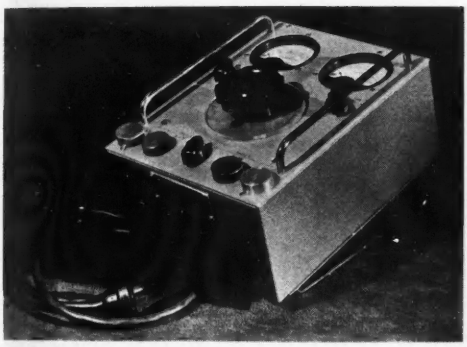
vertically in the side-walls and capable of easy removal without dismantling the furnace chamber. Control gear includes a voltage regulator for the high-temperature chamber and an energy regulator for the low-temperature chamber, which is fitted with heavy-section nickel-chromium wire heating coils. A temperature-indicating instrument is provided for each chamber ; alternatively, if desired, fully automatic temperature-control gear can be supplied.

A special feature is the doors, which, by means of a hinged arrangement, open in such a way that the steel outer face forms a charging plate, with the hot face turned downwards away from the operator.

The unit is supplied as a packaged unit (as illustrated) comprising the twin-chamber furnace, a control cubicle, a floor-mounting stand with open work-shelves, and a closed tool cupboard, but, as all items are separately constructed, they may be mounted in any desired location.

PORTABLE COMBINED METAL SORTER AND RESISTIVITY MEASURER

Designed for the sorting and identification of non-ferrous metals and alloys, the " Solartron " combined metal sorter and resistivity measurer, announced by the Solartron Electronic Group, Ltd., of Thames Ditton, Surrey, is a handy, portable, lightweight instrument, easy to read and to use. It is operated by dry batteries and may be used in open-air stockyards and scrap-metal yards, or where no electricity supply is available.



This portable metal sorter and resistivity meter, or test-set, consists of an investigating head on a lead, which is plugged into a small instrument case. The test heads are interchangeable and are designed to ensure that changes of ambient temperature do not cause drift, thereby affecting the readings of the instrument. The head is placed on the metal surface, which need not necessarily be flat, because measurement of the resistivity of the metal is not critically dependent on the spacing of the head from the surface of the metal under investigation. Identification is based upon the specific resistivity of the particular metal. Because this is a physical constant, of which every metal and alloy has its own specific value, its measurement identifies the metal being tested.

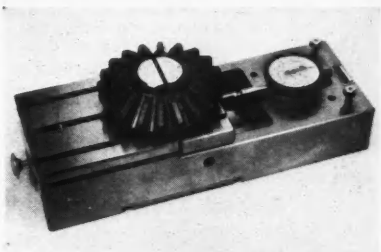
When the test head is brought near to the metal, its inductance changes. The oscillator in the instrument case ceases to oscillate, and the calibrated dial is rotated until the oscillator comes into action again. This point is indicated by the left-hand meter, and the reading of the calibrated dial at this point is the specific resistivity of the metal in micro-ohms/cm³. Provided that the metal or alloy has a specific resistivity between 1·0–28·0 micro-ohms/cm³, it may be measured, or, inversely, the metal or metals distinguished or sorted. Four metal standards are provided below the calibrated dial, by means of which the calibration of the instrument may be checked and the necessary correction made by the calibration control on the panel. A four-position switch and meter enables the high-tension, low-tension, and bias batteries to be tested from time to time.

The instrument can be used to test an extremely wide range of materials, including pure copper, gold, pure aluminium, molybdenum, radium, tungsten, cadmium, rhodium, zinc, brass, osmium, palladium, platinum, tin, tantalum, magnesium bronze, lead, and tellurium, and many of their various alloys.

MULTI-PURPOSE PRECISION BENCH COMPARATOR

Designed for the rapid and accurate checking of a large variety of internal and external forms, a new multi-purpose precision bench comparator, designated the " Ten-Fifty Check Master ", has recently been introduced by British Indicators Ltd., of St. Albans, Herts. The main body of the instrument is a rigid box-section casting of close-grained cast iron in which is mounted a friction-less transmission unit. The top table of the transmission unit and the top table of the comparator frame are ground in plane one with the other, and through both run three longitudinal tee slots. Anvils suitable for any given application are secured by means of these, the sensitive anvil being mounted on the table of the transmission unit, and the non-sensitive anvil on the table proper.

In use, the transmission is in contact with the dial indicator by means of an adjustable datum stop. Suitable gauging pressure is provided by a tension spring inside the body, attached to the transmission unit, provision being made for the reversal of this pressure when changing from external to internal checking. The sensitive anvil is retracted by means of the limit-knob, which restricts the travel of the transmission table in one direction. Interchanging of anvils is simply carried out, a pair of keeper pins being provided which anchor the sensitive table to prevent damage to the transmission unit. All anvils are tongued to locate in the tee-slots provided, and the location of the keeper pins facilitates approximate setting. The dial indicator is fully protected against the ingress of dust and oil by means of a spindle sheath and a perspex-faced shroud, which is quickly removable for the zero setting of the dial-gauge bezel.



The illustration shows the instrument set for checking the effective diameter of the splined bore of a bevel gear. Other applications covered by this instrument include the checking of internal gears, external splines, shallow external grooves, run-out and tooth engagement of gears, special external forms such as checking fir-tree and dovetail blade-roots, screw-thread effective diameters and pitches, and hemispherical inner tracks of ball-race shells.

ELECTRIC WRIST-WATCH

No larger than a conventional wrist-watch, yet stated to offer the highest accuracy and dependability ever achieved, a new electric wrist-watch, recently perfected by the Allied Products Division of the Hamilton Watch Company, of Lancaster, Pennsylvania, is operated by a miniature battery, instead of a mainspring. As a result, winding or periodic agitation is unnecessary, and the number of components has been reduced by about 33%.

Maxgrip has a unique friction driving ply which revolutionises a well known belt, improves pulley grip and gives

15% EXTRA POWER!

* This patented ply is made from special yarn, of cotton and rubber spun together under vacuum. Each cotton fibre is thus latex coated, ensuring a 50-50 rubber and cotton face throughout the life of the belt.

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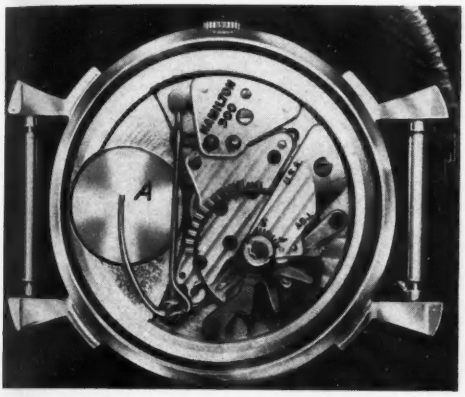
P.3821

The watch operates on chemical energy stored in the tiny gold-plated battery, which is about the size of a small shirt-button, but which is capable of powering the mechanism for a minimum of one year, corresponding to a power consumption of less than 0·00001 W. The energy from the battery (shown at *A* in the illustration) passes through a coil of fine wire fixed on a balance-wheel, which is caused to oscillate through interaction with permanent magnetic fields. The overall result is a precise miniature power-plant built into the balance-wheel, the essential difference between this type of motor and a conventional electric motor being that the new power-plant, combined with a balance-wheel, permits strict control of the flow of energy and enables the speed of the hands to be held to an accuracy of more than 99·995%.



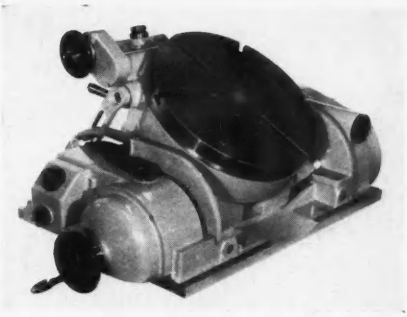
The coil is made of wire about 0·0004 in. in diameter, i.e., about one-fifth the diameter of a human hair, while the tiny magnets used in the electrical system are made of a platinum alloy (77% platinum, 23% cobalt) which is stated to have the highest energy content of any existing magnet. In this connection, each magnet weighs only 0·0056 oz, and is capable of supporting at least 215 times its own weight. These magnets, like the miniature battery (which is stated to be 400 times more efficient, in terms of space, than the mechanical energy stored in a mainspring), were specially developed for use in the new watch.

In addition to its exceptional accuracy, the Hamilton electric wrist-watch is claimed to be more nearly free from disturbance by stray magnetic fields than any other watch, so much so that those who work with or in proximity to magnets should be able to use it without interference.

OPTICAL ROTARY AND INCLINABLE INDEXING TABLE

A new addition to their existing range of optical rotary indexing tables is announced by Newall Group Sales Ltd., of Old Fletton, Peterborough. Believed to be the first equipment of its type to be marketed, the new model, available with either a 12- or 16-in. platen, is unique in that the scales for both rotary and inclinable movements are read directly to 1 sec of arc— a feature which should prove of great value in many branches of engineering industry where a considerable demand exists for a finer degree of accuracy in positioning workpieces with compound angles than has hitherto been necessary. To achieve this high degree of accuracy, the conventional eyepieces with 60-min graticules have been replaced by units embodying optical micrometers of the wedge-prism type, while reading at any position throughout the axis of inclination is facilitated by the provision of a unit containing eyepieces set at 90 deg. to each other for observation of the rotary scale.

Of contemporary design and robust construction to support heavy loads, the tables incorporate rotary and



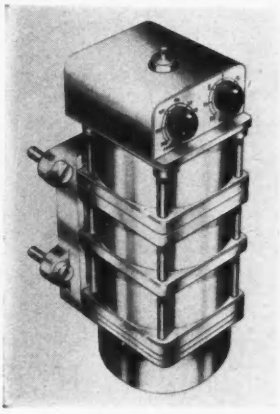
inclinable motions controlled by a worm drive to prevent slip when unlocked. Provided inside the inclinable base are locks which hold the work-table rigidly and without distortion ; generously proportioned support bars maintain the angle of inclination when locked. With both models, detachable datum blocks are supplied for location of the table square with the axis of inclination.

PNEUMATIC AUTOMATIC CONTROLLER

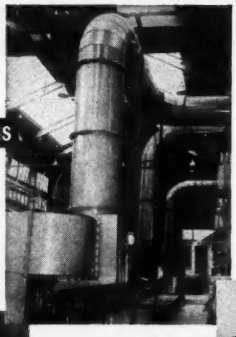
Built on the stack principle, but utilizing bellows, instead of diaphragms, the Cheltenham pneumatic automatic controller, announced by Cheltenham Auto Controls Limited, of Cheltenham, is an instrument which combines unusual stability with high sensitivity. Special care has been taken in the design of the various bellows units to give ample access for thorough cleansing after soldering, each unit being chemically cleaned before assembly. The controller provides proportional, plus integral, actions. Three proportional bands are available, i.e., 50, 100, and 200%. Integral time is delicately regulated by a high-precision needle valve with a needle taper of 1:1000. A separate derivative unit is supplied, where needed, this unit receiving and acting upon the original signal.

A novel feature is embodied in the nozzle unit, which permits the nozzle to be screwed in or out for zero setting, so that the adjustment is effected without moving the bellows stack and without upsetting spring rate in any way. The controller stack is arranged to "plug in" to an adaptor plate, to which it is secured by captive screws. Automatic shut-off valves in the adaptor plate ensure reliable isolation when the controller is removed. Adaptor plates are available for wall, panel, indicator, and recorder mounting. Restrictors are contained in the controller-stack assembly and are easily accessible.

Introduced with the controller is a wide range of relays, mainly bellows-operated. These relays deal with such functions as addition and subtraction, lead and/or lag, multiplication and division, variable ratio, etc. The booster relay in this series is capable of handling 7 cfm. Inlet and exhaust valves are independently adjustable, so as to permit selective sensitivity and zero level. The booster relay reacts to an input variation of 1·5 in. w.g.



COBEX Combats Corrosion



VICKERS-ARMSTRONGS

Leading from the anodising tanks at the Vickers Supermarine Works, South Marston, this exhaust ducting is made in COBEX by Horwitch Smith & Co. Ltd.

THE PROOF is in the organisations using it. The list is a long and impressive one. The following are just a few of the many world famous industrial companies which choose COBEX for important work involving corrosion resistance.

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A dish-ended vessel, fifteen feet long by eight feet diameter, being lined with COBEX/VELBEX LAMINATE for the storage of concentrated hydrochloric acid.

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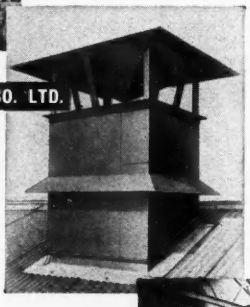

Cowl on the ducting system constructed entirely from COBEX in the Barry (Glamorgan) works of British Geon Ltd., a Distillers plastics company. Fabricated by Tanks and Linings Ltd., Droitwich.

ENGLISH ELECTRIC CO. LTD.

This famous organisation used COBEX (laminated with Velbex) for the ducting tank lining and facing panels in the Stafford electro-plating shop, equipped by W. Canning & Co. Ltd., Birmingham.

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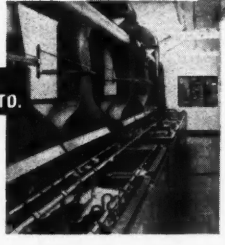

Fume ducting in new demonstration plating department at Woking, fabricated in COBEX by Prodorite Ltd., Staffs.

Write for leaflet and technical data to

BX PLASTICS LTD. *A subsidiary of the British Xylonite Co. Ltd.*, HIGHAM STATION AVENUE, LONDON, E.4. 'PHONE: LARKSWOOD 5511

NEWS OF THE MONTH

PERSONAL

Mr. John Stuart Bright, P.Eng., M.E.I.C., A.M.I.Mech.E., has been appointed General Manager of F. Perkins (Canada) Ltd., Toronto, a subsidiary of F. Perkins Ltd., the Peterborough diesel engine manufacturers. He succeeds **Mr. D. H. Milnes,** who has taken up an appointment outside the company.

Changes have taken place within the Birmingham firm of Birlec Ltd. A new Heating Division has been formed by the amalgamation of the Furnace and Induction Heating Divisions under the Management of **Mr. J. M. Paton. Mr. D. A. Hackett** has been appointed Sales Manager of the new division. A new Engineering Department has also been formed by the amalgamation of divisional drawing offices, under **Mr. G. H. Shirley** as Chief Engineer.

Mr. F. Eric L. Bache, M.C., LL.B., has been appointed Chairman of Geo. Salter & Co., Ltd., the West Bromwich spring and spring balance makers, in succession to his older brother, the late Mr. C. S. Bache.

Mr. E. G. Clarke, Managing Director of Acheson Colloids Limited, has been elected to the Board of Directors of Acheson Industries Inc., of New York, the parent Company to which the British unit of Acheson Colloids Limited is affiliated. It was also stated that Mr. Clarke has been appointed to the office of Vice President responsible for European Operations.

It is announced that **Mr. Frank Rowe, B.Sc., F.I.I.A.,** has been appointed a Director of The George Cohen 600 Group Limited—the holding company which controls the 600 Group. The appointment was effective from 1st January, 1957.

Two new sales appointments have just been announced by the Chloride Group of Companies. **Mr. T. Dawson** has taken up the new post of Group Export Sales Manager and will, in addition, be responsible for commercial liaison with the group's overseas associate companies. **Mr. P. C. Aspinall,** who was Assistant Export Manager under Mr. T. Dawson, becomes the new Export Manager for Chloride Batteries Limited in succession to Mr. Dawson.

The Ministry of Supply announces that **Mr. E. E. Haddon** has been appointed Director of Chemical Defence Research and Development in succession to **Mr. A. E. Childs,** who retired on 16th January, 1957.

On the retirement of **Mr. G. W. Preston, M.B.E., M.I.E.E.,** for health reasons, as General Manager, **Mr. R. B. F. Wylie, B.A. (Cantab.), B.L., LL.B.,** has been appointed to the new position of Director of the Copper Development Association.

Mr. V. S. Risoe, M.B.E., has been appointed to the Board of Associated Electrical Industries Overseas Ltd. and has resigned from the Boards of Associated Electrical Industries (India), Private Limited and Associated Electrical Industries Manufacturing Company, Private Limited.

Automatic Telephone & Electric Co. Ltd. announce that **Sir Alexander Roger** has retired from the Board with effect from 31st December, 1956, and has accepted the appointment of Honorary President of the Company. Sir Alexander Roger has been succeeded as Chairman of the Company by **Sir Thomas Eades,** who has been Managing Director for the past twenty years, and **Mr. C. O. Boyse** has been appointed Managing Director both with effect from 1st January, 1957.

Mr. C. de Rooy has been appointed Manager of Clarkson-Holland N.V. Amsterdam, and will be succeeding **Mr. A. Goddey,** who is returning to England to take another appointment with Clarkson (Engineers) Ltd., Nuneaton. Clarkson-Holland N.V. is a Manufacturing Company in Holland for Clarkson Autolock cutters, and also sole distributors throughout Holland of the full range of Clarkson Milling equipment.

Wild-Barfield Electric Furnaces Ltd. announce that as from 1st January, 1957, **Mr. J. E. Oram,** hitherto Director and General Manager, is appointed Managing Director of the Company. The composition of the Board remains unchanged, Mr. G. R. Barclay, O.B.E., continuing as Chairman and Dr. F. W. Haywood as Technical Director.

Thos. W. Ward Ltd. announce the appointment of six Local Directors from 1st January, 1957 : **Mr. Ernest Springthorpe,** who was Chief Draughtsman to the late Mr. James Bussey in the Constructional Steelwork Dept. **Mr. C. Norman Bradshaw,** who for several years has had important duties connected with personnel and administration. **Mr. Eric Wolstenholme,** who will be remembered in Sheffield on the scrap iron and steel side of the business and is now assistant to the London Director. **Mr. George Page,** Manager of the Export Dept. of the Company. **Mr. Reg. Hadfield,** Manager of the Foundry Supplies and Refractories side of the business. **Mr. Alfred Kissack** of the Machine Tool Section of the business. The new Local Directors have all spent their entire business life with the Company.

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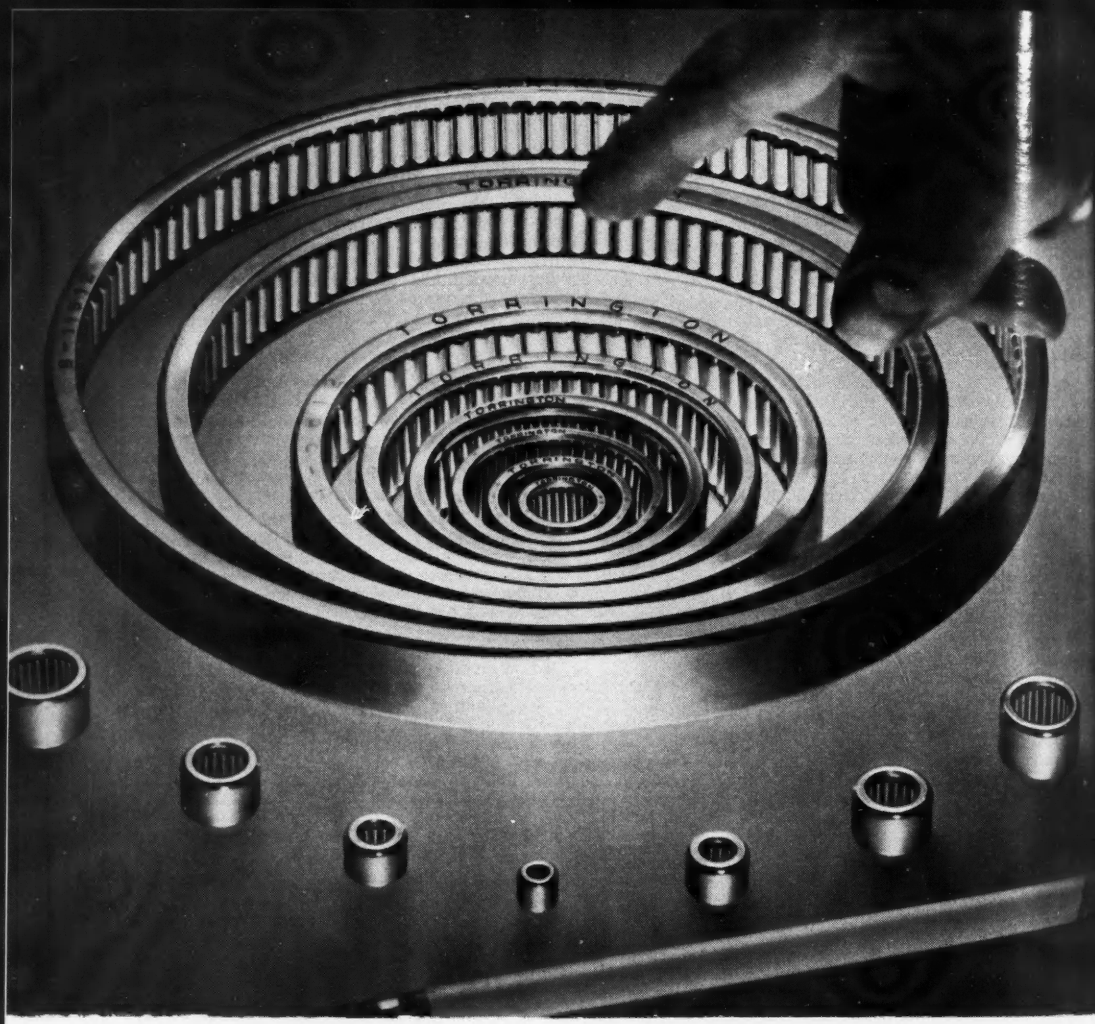

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Armour & Company Ltd., London, E.C.1, announce the appointment of **Mr. F. K. Johnson, B.Sc.,** to their Chemical Division, to handle the sale of cationic chemicals to the petroleum industry in the United Kingdom. Mr. Johnson was previously with the Production Department of the Associated Ethyl Company Limited.

Mr. J. T. Lommel has retired from Messrs. Arthur Lee & Sons Limited, Sheffield, at the age of 80. His position at the time was Export and Advertising Manager.

Mr. J. M. Brooke, M.I.M.E., Managing Director of the Venner Group of Companies at New Malden, Surrey, who manufacture time switches, electronic instruments, lightweight accumulators and Park-O-Meters, left this country on 12th January to investigate the possibility of setting up a company to manufacture Venner products in Australia. Mr. Brooke flew out by the North Pole route, reaching Sydney on 16th January to stay with H. Rowe & Co. (Pty) Ltd., the Venner distributor who will be assisting in the investigations. Following upon a visit to Jas. J. Niven & Co., the Venner distributor for New Zealand, Mr. Brooke will fly back via New York and is scheduled to return to London on 1st March.

The Board of the Brush Group Limited state that **Mr. Miles Beevor,** Deputy Chairman and Joint Managing Director of the Brush Group Limited has relinquished the post of Joint Managing Director. He remains as Deputy Chairman. **Mr. Ian T. Morrow,** Joint Managing Director, has been appointed Managing Director. **Mr. Charles Barnard,** hitherto Managing Director of Mirrlees, Bickerton and Day Limited, has been appointed to the Board of The Brush Group Limited with special responsibilities for the Group's Heavy Engine factories (Mirrlees, Bickerton & Day Limited, of Stockport, The National Gas & Oil Engine Co. Ltd. of Ashton-under-Lyne, and J. & H. McLaren Limited of Leeds). **Mr. M. C. Clear** has also been appointed to the Board of The Brush Group Limited, as Director in charge of Exports.

Mr. R. T. de Poix has been elected Chairman of the Zinc Development Association for 1957. Mr. de Poix is the Managing Director of Henry Gardner & Co. Ltd., and represents the Canadian zinc producers—the Consolidated Mining & Smelting Co. of Canada Ltd., and the Hudson Bay Mining & Smelting Co. Ltd.—on the Council of the Association.

The Minister of Supply, with the agreement of the Secretary of State for War, has appointed **Colonel G. C. Richards** Deputy Director of the Military Engineering Experimental Establishment, Christchurch. On taking up his new duties Colonel Richards was granted the acting rank of Brigadier.

Mr. F. P. Barribal and **Mr. J. A. Paul** have been appointed Directors of Mobil Oil Company Ltd. with effect from 1st January, 1957. Mr. Barribal has been Automotive General Sales Manager since 1952 and his portfolio as a Director will comprise all departments and sections concerned with the retail marketing of automotive fuels and lubricants. Mr. Paul, who has been Manager of Supply Department since 1953, will continue to manage that Department. In addition he will assume responsibility for the Wholesale Lubricating Oils and Foreign Shipping Departments.

At their meeting on Wednesday, the 23rd January, 1957, the Directors of John I. Thornycroft & Co. Limited appointed **Admiral Sir William G. Andrewes, K.B.E., C.B., D.S.O.,** to the Board.

Newall Group Sales Ltd. announce the appointment of **Mr. W. J. Vaughan** as technical sales representative for North-Eastern England.

It is announced that **Mr. Bruce Wilkinson,** formerly Managing Director of Electronic Tubes Limited, has joined The Plessey Company Limited as Divisional Manager of the Swindon Components Division.

The following appointments have been made to the Board of Imperial Chemical Industries Ltd. as from 24th January : **Dr. R. Beeching** becomes Technical Director ; **Dr. J. Ferguson** becomes Research Director ; **Mr. L. H. Williams** becomes Director in charge of Group B—Dyestuffs and Pharmaceuticals (effective from 1st April, 1957, on the retirement of **Mr. P. K. Standring**) ; **Mr. C. M. Wright** becomes Development Director. Two changes on the Main Board of I.C.I. are announced : **Mr. J. L. S. Steel** has been appointed Economic Planning Director, and **Mr. C. Paine,** previously Development Director, succeeds Mr. Steel as Group A (Heavy Chemicals) Director.

Mr. F. W. Cooper, B.Sc.(Eng), M.I.Mech.E., M.I.Prod.E., Principal of the Chance Technical College, Smethwick, has been appointed Education and Technical Officer to the Institution of Production Engineers. Mr. Cooper will take up his appointment in London on 1st April, 1957.

At a recent meeting of The British Power Press Manufacturers' Association the following officers were appointed for the new session: **Mr. Henry D. Challen, A.M.I.Mech.E.,** a Director of Messrs. Taylor & Challen Ltd., Chairman of the Association, **Mr. E. C. Seed, M.I.Mech.E., A.M.I.Min.E.,** Chief Engineer and Sales Manager of Messrs. Cowlishaw, Walker & Co. Ltd., Vice-Chairman of the Association. Elected as new Members of Council for the session 1956–7 were **Mr. T. H. Burton,** a Director of Messrs. Hordern, Mason & Edwards Ltd., and **Mr. C. E. Rockwell,** Managing Director of Rockwell Machine Tool Co. Ltd.

Mr. H. F. Akehurst has been appointed to the Board of British Insulated Callender's Cables Ltd. Effective from the 1st January he holds the executive position of Director (Overseas Operations).

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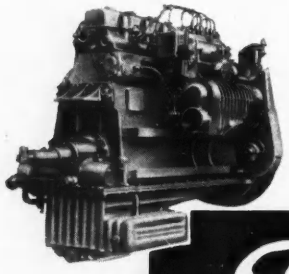

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BUSINESS NOTES

Remington Rand Ltd., 1–19 New Oxford Street, London, W.C.1, announce that a series of electronic computer courses are to be held during 1957. No previous electronic computer knowledge is necessary and details are available of special courses for Top Management, Senior Executives, Junior Executives, and Programmers, the time being split between London (plus Bristol, Manchester and Glasgow for Top Management) and Frankfurt, Germany. Special courses at the Univac European Computing Centre, Frankfurt will also be held in English.

The story of the first 50 years in the life of **The Churchill Machine Tool Company Ltd.**, Broadheath, Manchester, is told in a most attractive Golden Jubilee book. The progress from their Pendleton factory in 1906 to their modern Broadheath premises is well illustrated.

During the last week of 1956, and ahead of the scheduled dates, **Appleby-Frodingham Steel Company**, Scunthorpe, a branch of The United Steel Companies, Limited, brought into service two major items of plant—a new turbo-blower and a new open hearth furnace. The turbo-blower, which is twice the size of the four existing units at Appleby-Frodingham, is capable of delivering 150,000 cu ft of air per minute to the blast furnaces at a pressure of 35 lb per sq. in. ; the steam turbine generates 20,000 hp. Because of the additional blowing capacity now available, it will be possible to obtain a higher output of pig iron from the company's four blast furnaces. Final preparations throughout the Christmas holiday period enables the new 300/350-ton open hearth tilting furnace to be ready for its first charge on 27th December ; the furnace was tapped for the first time on the following day. The bulk of the 120,000 tons of steel produced annually in the new furnace will be supplied to the Steel, Peech and Tozer branch of United Steel, where the additional steel is required for the Brinsworth continuous medium strip mill, which is due to start up in mid-April. The 300/350-ton furnace is designed primarily for firing by cold coke oven gas and pitch creosote, but mixed blast furnace and coke oven gas can be used after minor modification. It is equipped with modern instrumentation and automatic control devices.

A new overseas development programme has been announced by **The Pyrene Company, Limited.** It includes the purchase of another Canadian subsidiary and the launching of a new manufacturing company in Australia. Brief details of the programme were disclosed recently when the Company announced the issue of 500,000 new ordinary shares. The proceeds of the new issue are estimated at about £320,000. Part of this sum is to be used for the purchase of C-O-Two Fire Equipment of Canada Ltd., at a price of 310,000 U.S. dollars. This is about half the price paid in 1954 for the existing Canadian subsidiary, The Pyrene Manufacturing Company of Canada, Ltd. A further sum is to be used in subscription of shares in a new Australian Pyrene company which is to be formed as a joint enterprise with Harrisons Ramsay Pty., Ltd., the Pyrene agents in that country. The balance of the new capital will be applied in the purchase of trade marks, patents and working drawings, and manufacturing and sales rights in various parts of the world.

In keeping with the continued expansion of **Wolf Electric Tools Ltd.**, and to offer an even more prompt, efficient repair service to Wolf portable electric tool users in the Home Counties, the Company has moved its Head Service Department from 239 Acton Lane, London, W.4, to larger premises. Tools in need of service should be addressed to Head Service Department, Newlon Works, off Stanlake Villas, Stanlake Road, Shepherds Bush, London, W.12. Tel. SHEpherds Bush 4414/5/6.

The Institution of Structural Engineers accepted, in 1954, an offer by the **Aluminium Development Association** of a Research Scholarship to enable the holder to undertake research on some aspect of the application of aluminium alloys to structures. It is expected that a report of the research work carried out under the first award of the Scholarship made in 1954 will be published towards the end of the present year. The conditions of this Scholarship have recently been revised and it is now open for award every third year for a two-year period, the value of the Scholarship being £600 per year. It is the intention of the Institution to make the next Award of this Scholarship in 1957 with a view to the successful applicant commencing his investigations at the beginning of the University Session in October next. Entries for the Scholarship to be awarded this year close on 31st March, 1957. The Scholarship is administered by the Institution and further particulars, together with forms of entry, should be obtained from the Secretary of the Institution of Structural Engineers, 11 Upper Belgrave Street, London, S.W.1.

The Belfast Branch Office of **Honeywell-Brown Ltd.**, has moved to much larger premises at 296 Albert Bridge Road, Belfast. The new office is fully equipped to deal with all inquiries for Industrial Instrumentation, Heating and Air Conditioning Controls, and Precision Switches.

Colt Ventilation Ltd., Surbiton, Surrey, announce that copies of the Paper on Condensation written by their Technical Director, Mr. L. Gordon Davies, have now been printed and that copies are now available free of charge on request to the company.

Vacu-Blast Limited, manufacturers of the well-known mobile dust-free shot blast machinery have established a company in France. The company is known as **Société Vacu-Blast (France)** and its address is 46 Rue Anatole France, Levallois-Perret (Seine), France.

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The A.E.I.-John Thompson Nuclear Energy Company Ltd. and **The Morgan Crucible Company Ltd.** are forming a joint company, **Nuclear Graphite Ltd.** This company will specialize in the machining of graphite blocks for the construction of the massive graphite moderator piles in atomic reactors—similar reactors to those in use at Calder Hall. The rapid expansion of atomic power generation demands a large increase in production capacity and it is now necessary to lay down additional plant. This will provide machining capacity to cover likely expansion in the British nuclear power programme and for overseas orders. A site is being prepared and preparatory work is already well advanced.

Two further regional stockists and agents have been appointed by **Compoflex Co. Ltd.,** 23–25 Northumberland Avenue, London, W.C.2, to handle Wyrem dust extraction and ventilation hose. Smith Bros. Asbestos Co. Ltd. (27 Cank Street, Leicester) will cover the counties of Leicester, Derby, Nottingham, Northampton and Rutland, and Rubberlast (Britain) Ltd. (131 Hunslet Road, Leeds 10) will deal with the East and West Ridings of Yorkshire. Both organizations will be able to supply Wyrem hose from stock, in bore sizes from two to eight inches and in lengths up to 30 ft. There are now six regional stockists, covering most of the country.

Leo Computers Ltd., makers of the famous LEO Automatic Office, have now completed their move to new premises of 30,000 sq ft at Minerva Road, North Acton. This change has been made necessary to meet the demand for their new model, LEO II, the prototype of which is now in operation at Cadby Hall. LEO II, like the earlier model, has been designed especially for clerical work. Whilst it incorporates the fundamental design of LEO I—the soundness of which has been proved by three years continuous operation on a large variety of clerical tasks and by five years on mathematical work—it includes many improvements evolved as a result of this practical experience.

B.S.A. Tools Limited are pleased to announce that they have been able to make arrangements with their associated company, B.G. Machinery Limited, for the servicing of B.S.A. Built Landis Grinding Machines. Users of B.S.A. Landis machines are asked to make servicing requests direct to **B.G. Machinery Limited,** Montgomery Street, Sparkbrook, Birmingham 11. Telephone No. VICtoria 2351—(9 lines).

Controlled stretched Duralumin plates, which were previously available only from the United States, are now being manufactured in this country by a special process developed by **James Booth & Co. Ltd.,** Argyle Street Works, Birmingham, 7. The company has recently supplied to a British aircraft constructor a number of these plates—some of which are believed to be the largest of their kind ever used in this country. The largest plates were 25 feet 4 inch long, 2 feet 8 inch wide and 2 inch thick, weighing 1,880 lb each, and, in spite of their size, had all been subjected to a controlled stretching operation to eliminate the internal stresses occasioned by heat treatment. This method of operation enables plates to be machined without subsequent distortion.

A selection of resistance welding machines and electronic equipment manufactured by **Sciaky Electric Welding Machines Ltd.,** Slough, will be exhibited in conjunction with the Midlands Electricity Board, at their Industrial Showrooms, 247 Chester Street, Aston, Birmingham, from 18th February to 2nd March, 1957. Open Monday to Friday, 10 a.m. to 12 noon and 2 p.m. to 4.30 p.m.

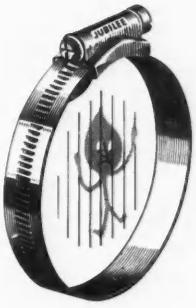
Air Control Installations Ltd. have opened a new branch office, Cross House, Westgate Road, Newcastle-on-Tyne 1. Telephone No. 28861.

The fourth display of instruments organized by the **Scientific Instrument Manufacturers' Association** was recently opened at their headquarters, SIMA House, 20 Queen Anne Street, London, W.1. The following members are showing a selection of their latest products. The period of the exhibition is three months. Advance Components Ltd., British American Optical Co., Chance-Brothers Ltd., Edwards High Vacuum Ltd., Ekco Electronics Ltd., Electronic Instruments Ltd., Ellis Optical Company, Foster Instrument Co. Ltd., A. Gallenkamp & Co. Ltd., Hendrey Relays Ltd., Hilger & Watts Ltd., Marconi Instruments Ltd., L. Oertling Ltd., W. F. Stanley & Co. Ltd., H. W. Sullivan Ltd., Unicam Instruments Ltd.

The Coventry Gauge & Tool Co. Ltd., Coventry. announced that their Midlands Area Technical Sales Representative, Mr. A. D. Barnsdale, relinquished his employment with them on 31st January, 1957. The appointment of another Technical Sales Representative for this area will be announced later and, in the meantime, technical queries and advice are being given direct to customers from Coventry.

Tecalemit Limited, Plymouth, Devon, lubrication equipment manufacturers, announce that they now have a new design of capillary fittings in production. These capillaries when used with their pumps, will ensure that bearings employed on machinery used for printing, glass-making, plastic moulding, machine tools, sheet metal and many other applications, are given the correct amount of oil where small quantities are required at regular or periodic intervals.

The Baldwin Instrument Co. Ltd., Dartford, Kent, announce that they have recently inaugurated a Fluid Power Division to control the sales and development of the Company's range of pneumatic and hydraulic equipment. Mr. D. A. Larner, A.M.I.Mech.E., is the Head of this new division.



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BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 2 Park Street, London, W.1)

Papers for electrical purposes. B.S. 698 : 1956. Because of the urgent need for a British Standard for insulating papers at the time, the first edition of B.S. 698 was published in 1936 without waiting for the results of some experimental work which was then being done by the Electrical Research Association. This work was interrupted by the war, but has since been completed, and the British Standard has now been revised in the light of the conclusions reached, and of a further study of the needs of the electrical industry. The requirements of the new edition are generally similar to those of B.S. 698 : 1936, but many changes in detail have been made and the opportunity has been taken to adopt wherever possible methods of test recommended by the Technical Section of the British Paper and Board Manufacturers' Association. The standard deals with a wide variety of papers, and for some properties it is necessary for the purchaser to state his requirements within the range specified in the standard. Price 7*s.* 6*d.*

Aluminium and steel-cored aluminium conductors for overhead power transmission purposes. B.S. 215 : 1956. In this revision of B.S. 215 : 1934 the standard has been divided into two separate self-contained parts, as follows : Part 1. Aluminium conductors for overhead power transmission purposes. Part 2. Steel-cord aluminium conductors for overhead power transmission purposes. Each part contains all the main provisions and details previously specified for the appropriate kind of conductor, but the number of standard sizes has been reduced and the reduced lists contain some sizes which did not appear in the 1934 edition. The amendment published in 1952 to provide for changes in manufacturing technique has been incorporated in this edition with some further modifications which have since been found desirable. A wrapping test replaces the elongation test for aluminium wires and the wrapping test for steel wires is now the same as that specified in B.S. 2763 for steel rope wire. The procedure for sampling and testing has been revised.

In previous editions of B.S. 215, the maximum resistance of each size of aluminium wire was specified. In this revision the maximum resistivity of the material is specified and is the standard value on which world agreement has been reached by the International Electrotechnical Commission. The values of other physical constants of aluminium and of steel core wire given in this edition have also been agreed or provisionally agreed by the I.E.C. Appendices give information on the strength of stranded and composite conductors and on modulus of elasticity and coefficient of expansion. A new feature in each part is a list of commercially accepted code names for use in ordering British Standard conductors. Price 3*s.* 6*d.* each part.

Tensile testing of metals. B.S. 18 : 1956. For fifty-two years this British Standard has been in constant use by industry. First introduced for the purpose of eliminating unnecessary variations in tensile testing requirements, it has since been revised four times to keep its requirements abreast of advancing technology. B.S.I. now announce the publication of a fifth revision. In preparing it, the aim has been to describe and define all procedures relating to tensile testing which are common practice throughout industry or—in certain instances—within large branches of industry.

Requirements are specified for test pieces for tubes. They relate to the testing of the tube whole, or when strips are cut longitudinally or circumferentially from it, or when the test piece is turned from the wall of a thick tube. Test pieces for wire are also dealt with. A section dealing with standard methods of procedure for tensile testing includes a revised clause on proof stress and new clauses on permanent set stress and proving tests; other clauses deal with yield, tensile strength and percentage elongation. Terms used in the tensile testing of metals are defined ; one of them is " tensile strength " which has been adopted in place of " ultimate tensile stress ". Detailed description or definition of procedure which is related to specific materials has been avoided, but the foreword lists—as a guide to committees engaged in the drafting or revising of standards for materials—points which may need to be separately specified for each material. An appendix gives a comparison between the symbols used in the standard and those adopted by ISO. Price 4*s.*

Fastening threads of B.S.P. sizes (formerly in B.S. 84 : 1940). B.S. 2779 : 1956. Originally specified in B.S. 84 (Screw threads of Whitworth form), requirements for the dimensions of these fastening threads have been transferred to this new British Standard to avoid misunderstandings as to their application. B.S. 2779 deals with fastening threads of B.S.P. nominal sizes from $\frac{1}{8}$ in. to 6 in. inclusive. " Close " and " Medium " class tolerances are specified for external and internal threads, " Free " class for external threads and " Normal " for internal threads. The tolerances on these threads are such as to make them unsuitable for joints where a pressure-tight seal is made by the making of the threads. Price 5*s.*

Cylindrical land steam boilers of welded construction (other than water-tube boilers). B.S. 2790 : 1956. This new British Standard meets the long-felt need for standard requirements for the construction of welded cylindrical steam boilers. During its preparation account was taken of recognized methods of construction of welded boilers : the results of experience gained in using this type of boiler were also incorporated. It applies to direct fixed and waste heat cylindrical and loco-type boilers of fusion-welded construction for land purposes. Price 12*s.* 6*d.*

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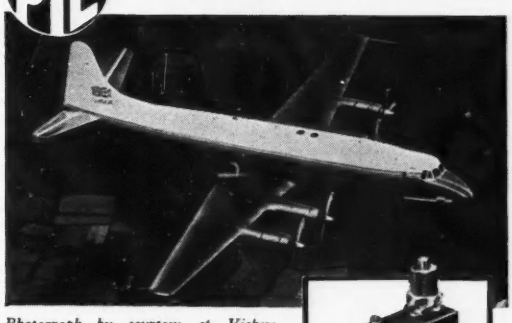

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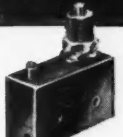

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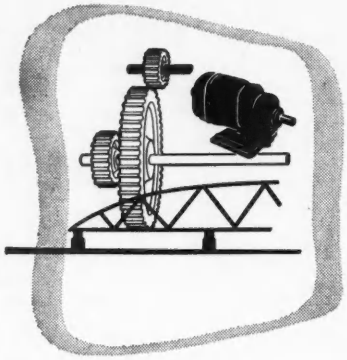

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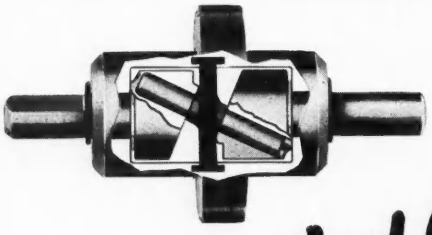

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CONTRACTS

The **English Electric Company Limited** has been awarded a contract to supply seven 750 hp diesel-electric locomotives to the **Midlands Railways of Western Australia.** The diesel engines and electrical equipment for these locomotives will be built at The English Electric Company's Preston Works, and the mechanical parts will be manufactured at the Company's Rocklea Works in Australia.

A £280,000 contract for marine diesels has been awarded to the **National Gas & Oil Engine Co. Ltd.,** Ashton-under-Lyne, a member of the Brush Group, by the **Polish Government Buying Agency** on behalf of the Polish Ship Salvage Co. of Gdynia. These engines will provide the propulsion and auxiliary power for two ocean-going tugs. Twin National B4AUM7 diesels with Hindmarch/MWD reverse reduction gearing developing 2880 shp at a propeller speed of 120 rpm and three National/Brush 150 kW sets at 600 rpm will provide the propulsion and auxiliary power in each of the vessels. Auxiliary power for emergencies will be provided by a McLaren M3 30 kW D.C. Generator and compressor set. **Messrs. Charles Hill and Sons of Bristol** will build the 1350 ton vessels which, with an overall length of 215 ft, will be among the largest diesel tugs in the world. Each tug will be fitted with ice strengthening at her bows for salvage work in the Baltic Sea and will also be suitable for tropical conditions. Delivery is due in 1958.

An order worth more than £3,000,000 has been placed by the **Central Electricity Authority** with **Metropolitan-Vickers Electrical Co. Ltd.** : it covers two 200-MW steam turbine-generator sets and associated equipment for Willington " B " power station, which will be situated adjacent to the River Trent, south of Derby. Commissioning dates are planned for 1961. The generators, which will be among the largest capacity machines yet built in this country, will have water-cooled stator windings. A Metropolitan-Vickers 30-MW generator, installed at the C.E.A.'s Bold power station several weeks ago, was the first large generator in the world to have this type of cooling.

The contribution made by **Birlec Limited** to research on aircraft and guided missiles is increased by an important order from the Ministry of Works for the **National Gas Turbine Establishment, Pyestock.** To be engaged in drying compressed air at the rate of 200 lb/second, the Birlec drying plant will form part of an altitude cell for testing jet engines at ground level while simulating the varying conditions between ground and high altitude.

The English Electric Company Limited has received an order from the **British Transport Commission** to supply the alternating current electric traction equipment for 112 motor coach trains. This order is among the first to be placed for this type of equipment and follows the announcement by British Railways that future electrification for Great Britain, with the exception of the Southern Region, will use the 25 kV alternating current system on the standard frequency of 50 cycles per second.

Czechoslovakia has ordered from Great Britain a complete plant costing approximately £600,000 for making chipboard or particle board by the Bartrev Process. The order was secured in Prague by Mr. G. H. Lowe, Managing Director of **International Plastics (East) Ltd.** and Mr. M. Culme-Seymour, Sales Director, of **International Plastics Ltd.**

The Central Electricity Authority has awarded a contract to **Mitchell Engineering Ltd.** for the installation of coal handling plant at the Authority's new power station at High Marnham, Nottinghamshire. The contract is valued at more than £½-million.

BOOKS RECEIVED

The Plastic Methods of Structural Analysis. By B. G. Neal, M.A., Ph.D.(Cantab.), A.M.I.C.E. 364 pp., 85 illustrations. Publishers: Chapman & Hall Ltd., 37 Essex Street, London, W.C.2. Price: 45/–.

The plastic methods of structural analysis have been developed for the purpose of calculating the plastic collapse loads for frames, with the ultimate objective of establishing a rational economical design procedure. These methods rest on certain basic assumptions concerning the behaviour of structural members in flexure which are closely obeyed by mild steel, and may also prove to be obeyed by other ductile materials. Quite apart from their applicability to mild-steel frames, the plastic methods are of considerable intrinsic interest, since they represent one analytical extreme in which conditions of collapse are discussed, whereas the elastic methods represent the other extreme in which the entire structure obeys Hooke's Law. Furthermore, the plastic methods of analysis may now be said to be as fully developed as the elastic methods.

This authoritative work by Professor Neal will undoubtedly be of considerable value, not only because of its interest to practising structural engineers and research workers, but also because a study of the plastic methods and their contrast with the elastic methods of analysis can constitute an important addition to an undergraduate teaching curriculum. The first four chapters are concerned solely with a presentation of the plastic methods of analysis, and the material contained therein could profitably be included in the later stages of an undergraduate course in the Theory of Structures. The remaining four chapters deal with topics which are closely associated with the plastic methods, and are suitable for more advanced study, although selected material from these chapters should be included in an undergraduate course. To assist in the use of the book, examples have been provided at the end of each chapter.

Circuit Theory and Design. By John L. Stewart. 494 pp., 465 illustrations. Publishers: John Wiley & Sons, Inc., New York, and Chapman & Hall Ltd., 37 Essex Street, London, W.C.2. Price: 76/–.

Avoiding abstract mathematics, this informative new work applies modern network theory to the understanding of vacuum tubes and feedback systems. In doing so it assembles a great deal of information which has previously been scattered throughout the literature in relatively incomplete form.

Pole-zero design methods, founded on an easily grasped pictorial representation, are employed extensively. They are used in developing design methods for a great variety of circuits, both with and without vacuum tubes, and for systems with and without feedback. Included are examples relating to practical system design.

The presentation of feedback amplifiers, oscillators, and servomechanisms is unique. In discussing feedback systems, Professor Stewart emphasizes the precision realization of a prescribed closed-loop transfer function. Also, many topics which are especially useful adjuncts to circuit design are given attention, e.g., function design, normalization, and the use of ideal transformers.

Applied Electrical Measurements. By Isaac F. Kinnard, D.Sc. 612 pp., 369 illustrations. Publishers: John Wiley & Sons, Inc., New York, and Chapman & Hall Ltd., 37 Essex Street, London, W.C.2., Price: 120/–.

This useful reference book is intended to cover broadly the basic principles of commonly employed electrical measurement devices and their application to the measurement of electrical and non-electrical quantities. Sufficient theory is given for an intelligent application of the principles involved to be made, and liberal references are included, enabling those who wish to do so to make a more thorough study. As a result, this volume will be useful not only to students but also to those scientists, engineers, and technicians who are faced with measurement problems.

The subject matter is treated within the framework of the familiar branches of physics, i.e., electricity; light; heat; statics and kinetics; liquids and gases; and time. To ensure the usefulness of the material, it has been chosen by men whose job it is to apply the principles of electrical measurements in industrial practice. It is obviously impossible to include every detail that might be needed by the individual worker; hence, sources of additional information appear in a large but selective bibliography.

In addition to the information on theory and applications, a method is shown for analysing and synthesising measurement devices and systems. This method gives a logical " feel " for the subject and thus aids overall understanding in this very broad area of study.

Digital Calculating Machines. By G. A. Montgomerie, B.Sc., A.M.I.E.E., A.M.I.Mech.E., Assoc.I.R.E. 266 pp., 146 illustrations. Publishers : Blackie & Son Limited, 17 Stanhope Street, Glasgow, C.4. Price : 30/-.

During the past ten or fifteen years, there has been a marked increase in the use of calculating machines of all kinds, particularly for scientific and engineering purposes, and more and more people now have to use and to be familiar with them. This book, then, serves a very useful purpose in providing a general introduction to digital calculating machines for scientific and engineering users, surveying the whole field and containing numerous illustrations of equipment, together with detailed examples of how it is used.

Chapters in this interesting and well-produced work deal with key-responsive adding machines, key-set adding machines, four-rule machines (lever-set, hand-operated ; key-set, electrically operated ; and automatic), computing with four-rule machines, punched-card machines, special-purpose machines, universal sequence-controlled machines, electronic sequence-controlled machines, and programming of sequence-controlled machines.

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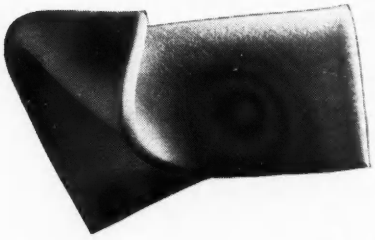

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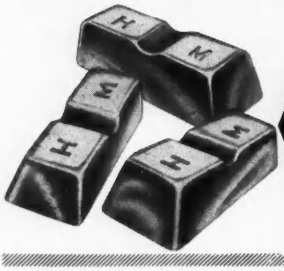

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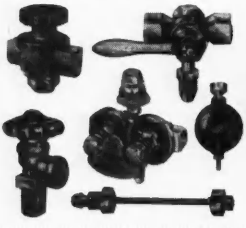

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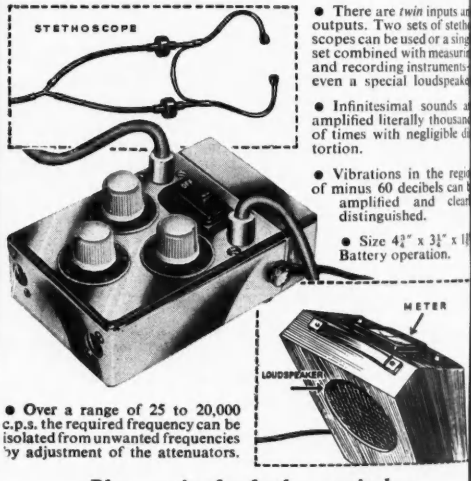

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1. Precision Gauge Cabinet Projector. Providing magnification factors of 10X, 20X, 25X and 50X, the new instrument described in this 2-page illustrated leaflet has just been added to the manufacturer's existing range of gauge and profile projectors. It has been designed to meet the ever increasing need for simple but highly accurate projectors for repetition or production inspection applications, and can be supplied with a plain base, a simple focusing stage, or fitted with a co-ordinate workstage for direct measurement. Although the cost is low, it is fitted with gauge lenses and a specially worked mirror which, together with the precision collimation system, ensure the same optical accuracy found in more expensive projectors.

The instrument can be used in either horizontal or vertical positions and provision is made for the insertion of a light filter for use when the 10 in. × 8 in. glass screen is being viewed continuously.

2. Extrusion Die Metal. This 4-page leaflet describes the advantages and performance of a die-metal recommended for the hot extrusion of non-ferrous metals, particularly copper or aluminium alloys. The material softens little when heated even to 1000°C and it is unaffected by prolonged heating at this temperature. Dies are manufactured by investment casting with round, hexagon, square and rectangular bores or with special profiles. Blank dies are also available, the spark machining process being employed for cutting simple profiles. This material is claimed to have many advantages over tool steel, and several examples are provided. Three grades of metal are available.

3. Electronic Comparator. A considerable amount of engineering detail is provided in the 8-page, well illustrated, brochure describing this comparator type of instrument. Consisting of a probe unit and an amplifier/indicator unit, measurements can be made to an accuracy of 0·00001 in. The operating pressure of the probe is less than 2 oz. thus ensuring safety of the work surface and long life to the probe tip. Four measuring ranges are provided, the centre-zero moving-coil meter being calibrated 0·005–0–0·005 in., 0·0005–0–0·0005 in., 0·12–0–0·12 mm, and 0·012–0–0·012 mm. The design of the probe is shown to eliminate friction and backlash, while the probe tips can be replaced. Additional current and voltage outputs are available from the amplifier to enable other recording instruments to be used in conjunction with the instrument. For high accuracy comparator measurements a comparator stand is described, having a vertical capacity of 6 in., a throat depth of 3½ in. and a hardened steel worktable.

4. Low-Flow Oil Meter. This literature provides a complete technical description and specification of a new meter intended for flows up to 200 gallons per hour and for installation in pipe lines of between ⅜ in. and ¾ in. Being of compact design and giving an accuracy of within ±2% from 1·5 to 200 gallons per hour it is ideal for recording the fuel consumption of small oil-fired furnaces. A direct reading counter, together with a circular scale fractional pointer-indicator, has a maximum registration of 1 million gallons or 10,000 m³. The calibration may be in either gallons or metric units. A maximum pressure of 100 lb and a maximum temperature of 200°F are permissible under working conditions. Operating data at various viscosities are given together with fitting and dimensional sizes. An outstanding factor in the design of the meter is the ability to register accurately very low flow rates while retaining excellent flow performance.

5. Automatic Press Guard. Designed as an improvement on the basic interlock guard system, this new equipment provides sequential operation ensuring that the press cannot be operated until the guard has been closed. The sequence is controlled by one depression of a foot pedal operating the pneumatic control of the guard. Upon the guard reaching the closed position it actuates a valve to pass air to a second cylinder to depress the clutch pedal. The time duration of the various actions can be set precisely and the whole equipment adjusted to provide automatic cycling of the complete operation. The 8-page brochure is well illustrated and describes the available models together with the increased productivity arising from their use.

6. Midget Assembly Trays. This leaflet illustrates the latest addition to a range of bench assembly and storage trays. Constructed for lateral interlocking in either a straight or a curved line, they measure only 2⅛ in. length, 2 1/16 in. width and ⅜ in. depth. Five colours are available and can be used to advantage for easy and rapid identification of the contents on the assembly bench.

7. Diamond Dressing Tools. Three types of diamond dressers are described in this 6-page, well produced and informative leaflet. Constructed from natural diamonds a profiled polished face is formed in such a manner that the cutting edges lie in the hard grain zones of the diamond. The resistance to wear of these hard grains is up to ten times that of the soft, and a special indexed holder ensures that the diamond is automatically brought into the correct cutting position. In addition to the profiled single pyramid diamond, two multi-pyramid types with wide and narrow pitches are available. The various holders and bits for use with the diamonds are illustrated and specified.

8. High Conductivity Copper Alloys. This is a 54-page illustrated booklet dealing with four copper alloys—Cadmium copper, Chromium copper, Silver copper and Tellurium copper. These alloys are of importance in the electrical industry where high electrical and thermal conductivities are required. The composition, production, physical and mechanical properties, etc., are discussed. There are twenty-five useful tables plus equilibrium diagrams, and photographs which illustrate some typical applications of these alloys.

9. Cutting Fluid Additive. The advantages and increased production rates obtained when using this additive are illustrated with photographs of typical machined work and high-magnification photographs of metal surfaces in this 10-page brochure. The chemical action of the additive is to rapidly produce a surface effect in direct proportion to the intensity of the heat, and which will prevent welding of the tool and the work-piece. It can be used with either oil or water based fluids and, even when used with water, it assures complete rust protection.

10. Epoxide Resin Paints. The contents of this small 8-page booklet provide much useful information on the subject of finishes based on epoxide resins which are now assuming great importance due to their high chemical resistance and adhesion qualities. Both stoving and air drying paints are described, the former for high or low temperature work, and the latter of spraying or brushing quality in decorative and high chemical resistance (Cold Cure) types. The outstanding features of each type are described and reference is made to many of the applications for which they are most suitable. Drying times, temperatures, preparation, application, storage, etc., are also covered.

11. Rolling Gear Testers. The equipment described in this 32-page catalogue is designed for the measurement, to required limits, of the variations of centre distance when a gear under test is rotated, under spring pressure, against a master gear. This double flank or composite test is quick and practical and, in the case of fine pitch gears, the only method available apart from projection. The common gear errors of eccentricity, form spacing and backlash are all indicated. Two models for use in the standard room and precision gear shop are first described. One is for fine pitch gears between 20 and 120 D.P. and between 0·1 in. and 5 in. diameter. For the larger gear systems a second model covers from 4 to 40 D.P. and between 1 in. and 14 in. diameter, and is intended for the automobile, machine tool and aircraft industries. For general workshop use with the more common type of production gear, a simple and low cost model is described for gears up to 14 in. diameter and between 4 and 20 D.P. The requirements for the larger gears up to 24 in. are met with a fourth model. A range of attachments to extend the usefulness of these testers are illustrated and specified, and include a power drive for master gears, master gears and worms and a backlash calculator.

12. Nickel Plating For Engineers. This new 72-page publication has been written primarily for engineers, designers and others who are interested in the possibilities which plating offers in design. No attempt has been made to cover plating procedures in great detail, but practices have been dealt with in so far as they affect the properties of the resultant deposits. The text includes a description of surface preparation in general and for zinc-base alloys, aluminium and nickel-chromium-iron alloys in particular. Commonly used solutions, relevant plating proceedures to maintain high quality work, plant required, mechanical properties of deposits and methods of testing deposits are all dealt with in a highly compressed text. A good deal of information is given in the form of graphs or tables for quick reference.

13. Continuous Cast Copper Base Alloys. Standard tin bronze alloys not capable of being hot or cold worked into extruded, rolled or drawn bars, tubes and shapes, have previously only been available in short lengths as sand, permanent mould, or centrifugal castings. This attractive, 8-page brochure, describes a new plant that has been installed in the United Kingdom for the continuous casting of a wide range of alloys in a form equivalent in all respects to extruded, rolled or drawn bars and shapes. Standard lengths up to 12 ft, or longer for special requirements, can be manufactured in an alloy where the hard and soft spots are eliminated, and where the surface or internal abrasives, which would normally dull tools or reduce cutting speeds, are no longer a problem. Tables of comparison are used to indicate the superiority of the continuous casting process over sand and chill cast methods. The figures show better fatigue characteristics, and greater strength for impact, tensile and yield measurements and hardness tests. Castings can be made in rod form from 7/16 in. to 5 in. and in tube form from 1 in. to 5 in. o.d. with a minimum i.d. of 15/32 in. The dimensional tolerances for rod and tube are +0·004 in. to −0·006 in. and concentricity within 1½% of the wall thickness.

14. Laboratory Electromagnet. A 2-page leaflet describes this high-performance electromagnet which has standard pole-pieces of 100 mm diameter designed for nuclear resonance and induction measurements, the determination of magnetic susceptibility, adiabatic demagnetisation and microwave spectroscopy. The equipment which can be supplied complete with the necessary stabilized power supply units is of special interest to Universities and large research units interested in molecular structures. Other magnets are also available having a maximum field strength up to 35,000 gauss. Angle or conical pole tips are available. The air-cooled magnet coils in parallel have a continuous rating of 8A but this can be exceeded for short periods.

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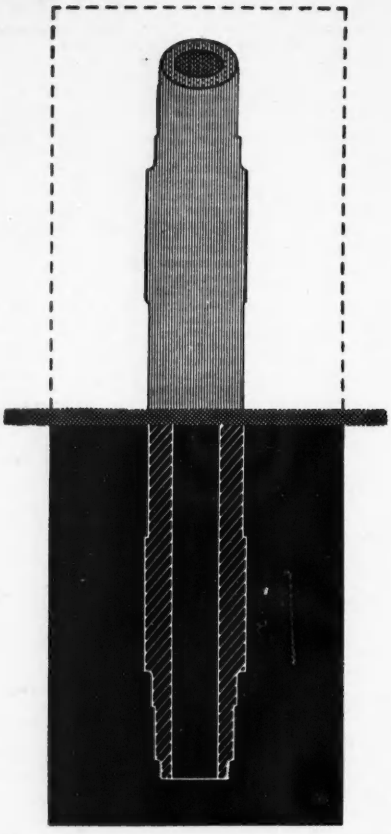

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